Skrotzki
& Vopat

POWER STATION
ENGINEERING AND ECONOMY

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POWER STATION ENGINEERING AND ECONOMY

(Second edition of APPLIED ENERGY CONVERSION)

Bernhardt G. A. Skrotzki, M.E., B.S.E.E.

ASSOCIATE EDITOR, *Power;*
MEMBER, AMERICAN SOCIETY OF MECHANICAL ENGINEERS,
AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS, AND
AMERICAN NUCLEAR SOCIETY;
REGISTERED PROFESSIONAL ENGINEER, NEW YORK STATE

William A. Vopat, M.S.E., M.E.

PROFESSOR OF MECHANICAL ENGINEERING,
AND HEAD OF DEPARTMENT,
THE COOPER UNION SCHOOL OF ENGINEERING;
MEMBER, AMERICAN SOCIETY OF MECHANICAL ENGINEERS AND
AMERICAN SOCIETY FOR ENGINEERING EDUCATION;
REGISTERED PROFESSIONAL ENGINEER, NEW YORK STATE

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SECCION CENTRAL HIDRAULICA
U.T.E.
RINCON DEL BONETE

PREFACE

This second edition of "Applied Energy Conversion" represents essentially a new book; it is retitled as "Power Station Engineering and Economy" to describe more pointedly the scope of the text. This new edition retains the essential aim of the first edition in emphasizing principles of operation rather than minute details of design of equipment. Power-apparatus design can be better learned in the actual practice of equipment building, in plant-design engineering groups, and in operating and maintenance organizations.

Major progress in the field of power engineering since the first edition required updating the book. In surveying the users of the first edition, we found a demand for more depth and wider scope in outlining the elements of a power-generating system as represented by the wide variety of today's power plants. Accordingly the book has been reorganized and thoroughly rewritten.

As in the first edition, this new book examines the power plant from the raw energy input to the rotating-shaft energy output. It follows the well-established tradition of covering the area of responsibility of the mechanical engineer. The equally important electrical aspects of the power plant have been left for other texts.

The rapid growth of our population has led to the development of a wide variety of systems to supply a number of urgently needed services. The power plant is probably the first device in our industrial civilization to work as an automatic integrated system of interlocked elements.

This book assumes that the reader has a background knowledge of the basic engineering sciences of thermodynamics, heat transfer, and mechanics. In dealing with the power-plant system the text has been divided into five major parts:

I. Energy processing
II. Steam power plants
III. Internal-combustion plants
IV. Hydro and meteorological plants
V. Power economics

Part I covers the basic practical elements of energy sources, power

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cycles, heat transfer, and combustion. These basic sciences form the working foundation of the modern thermal power station.

Part II covers the main and auxiliary elements of the steam power plant and their integration. Chapters on individual types of equipment aid in understanding their operating principles, which leads to an appreciation of the behavior and limitations of the complete power-plant system. This part includes a chapter on the rapidly developing field of a wide variety of nuclear-fueled power plants. Today nuclear plants are not of practical importance, except in military applications, but they undoubtedly will be playing an important civilian role in another generation. The forward-looking engineer educates himself early in a growing technology.

Part III discusses the two types of internal-combustion stations: (1) reciprocating engine and (2) gas turbine. Though the total capacity of i-c stations is only about 2 per cent of the total in the United States, they are important numerically, being of the same order as steam stations. Continuing development of these engine types attests the need of the power engineer to keep abreast of the field.

Part IV covers hydro and meteorological generating plants. From a practical standpoint the former is the most important and ancient type of this group in producing mechanical shaft power. The concepts of solar energy, tidal power, wave power, wind power, and terrestrial-heat power plants are interesting though of minor importance now. With our diminishing reserves of fossil fuels these concepts can be of crucial importance in the next century.

Part V, on Power Economics, remains essentially as in the first edition. We believe it still is the only comprehensive discussion of introductory engineering economics applied to power generation to have been published. A brief discussion of the probability theory as applied to determining optimum power-system capacity has been added, since this method is finding increasing acceptance in practice. This area of the study of energy supply should be enlarged in our engineering curricula because of its urgent importance in professional engineering practice.

Recognizing the importance of opportunities to apply newly learned principles, the book continues to include an ample number of problems at the end of each chapter.

The book has been written for the needs of courses of study in power plants for junior and senior engineering students. In addition, we believe that many practicing power engineers and plant operators will find the text valuable as a working reference in power-plant theory and application.

We are indebted to many friends, students, and associates for their

comments, which helped to shape what we believe to be a better and more useful book. The authors are particularly grateful to Professor Walter S. Kut for his aid in reviewing Part III and to the mechanical engineering department staff at Cooper Union for their willing aid in preparing the manuscript.

BERNHARDT G. A. SKROTZKI
WILLIAM A. VOPAT

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PART I

ENERGY PROCESSING

SECCION CENTRAL HIDRAULICA
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CHAPTER 1

RAW ENERGY

1-1. Importance of Energy. A most casual look at our civilization shows the important part played by the supply and control of energy. Many functions necessary to present-day living grind to a halt when the supply of energy stops. While electric power is a common form of energy used in the home today, there are other forms used industrially and domestically. Energy may be sold like other commodities at so much per unit quantity.

Electrical and mechanical energy are manufactured commodities, just like tangible items such as clothing, furniture, or tools. All manufacturing activity processes raw materials as found in nature into forms more useful for human use. Production of electrical and mechanical energy is similar, entailing the processing of natural raw forms of energy into more useful forms. It differs in one important respect: other manufactures may be processed at will and consumed as needed, but energy must be processed from the raw form and transmitted to the point of use at the instant it is needed. The entire process takes only a fraction of a second. This instantaneous production introduces technical, economical, and political considerations unique to the electric-power industry.

Raw energy in falling water, in deposits of coal, oil, and gas, and in certain types of refuse has yielded most readily to control and is applied commercially today. The amount of energy that may be extracted from a given source depends on its nature and the way it will be used. When it is used for space heating, a large portion of the raw energy may be converted to heat; however, when it is used to perform mechanical work, the laws of thermodynamics limit the amount that may be so converted.

Despite the many practical forms of energy conversion from fuels and water power persistent efforts are being made to extract energy from the winds, solar radiation, and the fissioning of uranium. Such conversions have been experimentally realized, with uranium showing the most promise. Winds have been used for many years as a source of mechanical energy, but their use for electric generation in central-station systems is still an experimental subject.

1-2. Energy Consumption. As technology advances, the consumption of energy steadily rises. Figure 1-1 shows the energy consumed annually in the United States since 1800 and the form in which the energy was supplied. In 1800 most energy was used for heating, supplied by burning wood. A minute amount of hydro power supplied shaftwork for milling and other needs. Coal supplied only about ½ per cent of the heating energy. Power engines were unknown except for some crude forms of steam engines.

Wood supplied the major energy share up to about 1890. Coal use began growing as an energy source about 1850. Oil and natural-gas use began growing after the Civil War. Coal reached its peak consumption during the two world wars. Oil and natural gas supply the major share of energy today.

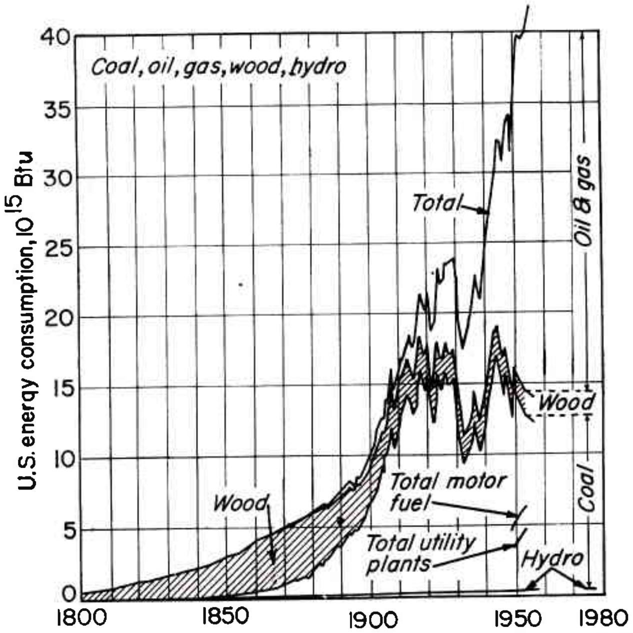


FIG. 1-1. Energy consumption in the United States increased many fold since 1800. Central stations use little more than 10 per cent of total.

Hydro generating stations supply the smallest share of energy today, on the order of 1 per cent. Wood still plays a significant part in supplying energy, mostly in the form of wood wastes.

To give scale as to the use of all this energy, two curves in Fig. 1-1, for the years 1949 to 1954, show the amount used as motor fuels and as fuels for central generating stations. The motor fuels—gasoline and diesel oil—power America's automobiles, buses, and trucks. The central-station fuels include coal, oil, gas, and wood.

Figure 1-2 shows the trend of total energy consumption in the United States, comparable with Fig. 1-1, on a semilogarithmic plot. The dotted line superimposed on the lower curve shows a long-term conservative trend in annual consumption. The dash-dot line shows the optimistic trend drawn through the peaks of energy consumption. Based on the latter, energy consumption in 1980 will be about 50 per cent more than in 1955, but conservatively only about 20 per cent more.

The upper curve of Fig. 1-2 shows the annual per capita consumption of energy in the United States. Increasing mechanization since the start of the twentieth century shows a rising trend, while the nineteenth century stayed at a nearly constant level.

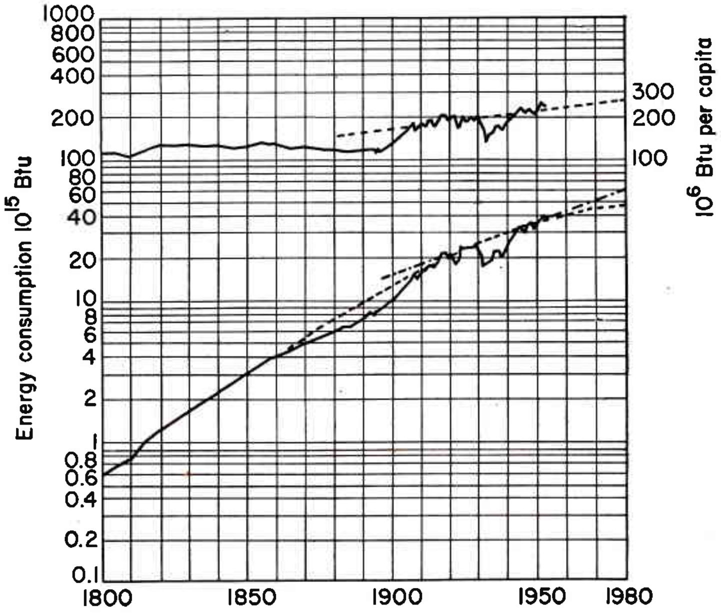


FIG. 1-2. Long-term trend of energy consumption in the United States shows decrease in the *rate* of growth, lower curve. Upper curve shows rising rate of energy use per capita in the United States.

1-3. Central-station Energy. We are primarily concerned with central-station generation of electricity, which consumes only about 10 per cent of the total energy used annually in the United States. This industry started only a few years before 1900 and has expanded rapidly ever since. Figure 1-3 shows the installed capacity of central stations since 1900 and an optimistic forecast of future installations.

The amount of energy produced annually by a station depends on how hard it is worked. Since there are 8,760 hr in a year, each kilowatt of capacity could theoretically produce 8,760 kwhr in 1 year. But since demands for electricity vary from hour to hour, the total capacity of a station is used only for a few hours in any one year. On the average a kilowatt of capacity produces much less than 8,760 kwhr. Figure 1-4 shows the annual generation per kilowatt of capacity since 1900. The rising trend of this curve reflects the increasing number of uses found for electrical energy.

Central-station electrical generation is only an intermediate step in the supply of electricity for a wide variety of uses. Figures 1-5 and 1-6 show a schematic suggestion of the mining, conversion, and transmission of energy to the ultimate consumer. Mineral fuels must be removed from underground strata by mining or pumping, then cleaned or refined and transported to the generating station. Here the fuel gives up its energy for conversion to mechanical energy and then electrical energy. Finally, in electrical form the energy travels to the ultimate consumer over established circuits.

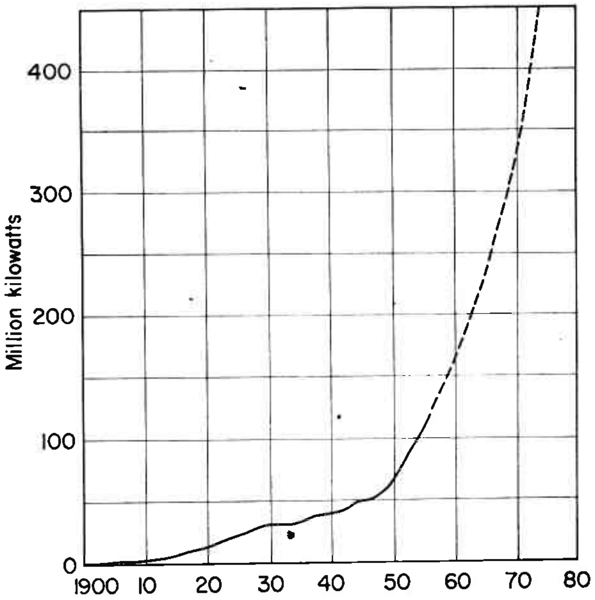


FIG. 1-3. Central-station installed capacity showed steady growth since 1900. Future demands for electrical energy indicate need for doubling capacity every 8½ to 10 years in the future.

The central-station designer must assure himself of having enough fuel to meet the needs for the life of the plant. He also must place his plant so that fuel-transportation and electric-transmission costs are balanced to get a minimum over-all cost. These factors must be considered, together with many others that we shall meet in this book.

A power plant is a complex assembly of equipment. For a long time designers have dreamed and hoped for simpler methods of converting fuel energy into electricity without the intermediate steps of heat transfer and production of mechanical energy. Thermocouples seem to offer a glimmer of hope, but to date they have not produced more than minute amounts of electrical energy. Fuel cells now being investigated in laboratories seem to offer some promise. Fuels burned in special cells under controlled conditions develop ionized particles. These can be collected at electrodes to produce an electric voltage and sizable currents. Much research still needs to be done in this area.

Nuclear energy has fired the imagination of power producers because of the tremendous amount of uranium that seems to be available. By 1960 there will be nearly a million kilowatts' capacity of nuclear-powered central stations producing electricity in the United States. These are all experimental, however, because electrical energy produced by these plants will cost several times that generated by conventional fossil-fuel-fired plants. Concentrated development efforts probably will make uranium-fueled plants competitive in the future.

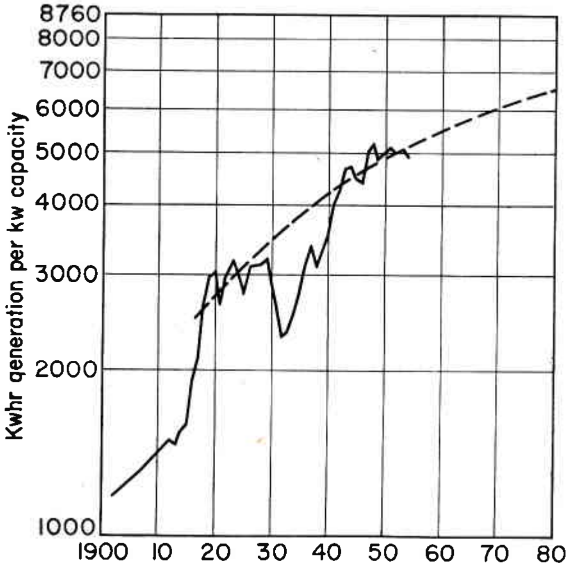
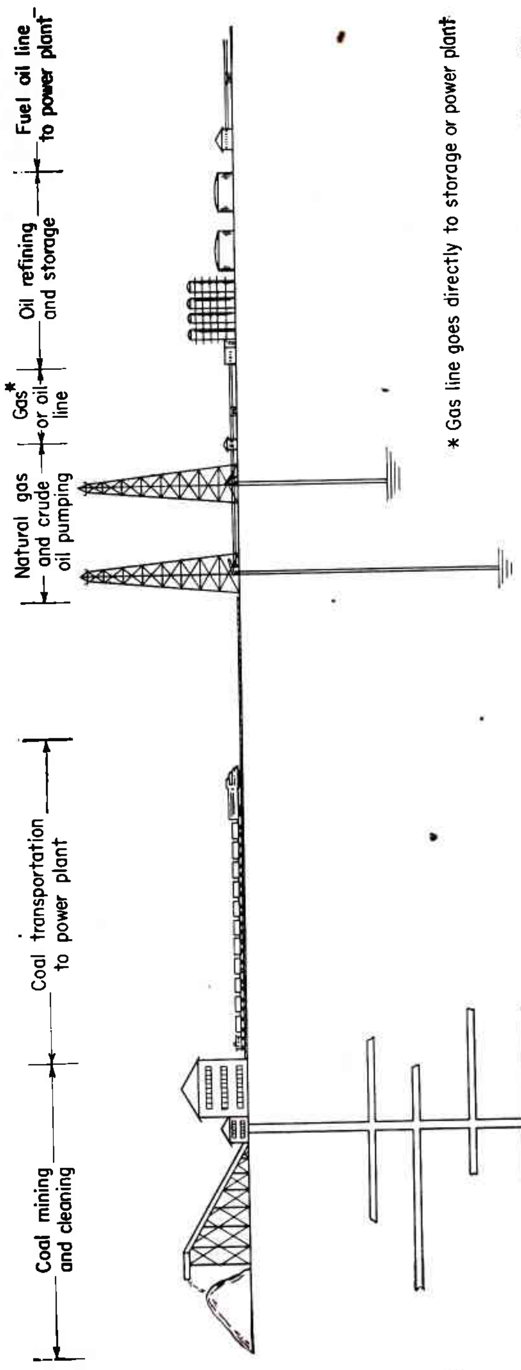


FIG. 1-4. Increasing need for electrical energy through more hours of the day causes 1 kwhr of central-station capacity to produce more kilowatthours in 1 year.

Within the last few years the fusion bomb achieved the conversion of matter to energy by fusing the nuclei of heavy hydrogen and tritium to release enormous amounts of energy. Scientists dream today of applying this process to practical power production. Our sun and stars use this process to produce their almost unlimited energy.

Heavy hydrogen exists in sea water at the ratio of 1 in 6,000 hydrogen


Coal mining and cleaning
Coal transportation to power plant
Natural gas and crude oil pumping
Gas* or oil line
Oil refining and storage
Fuel oil line to power plant*

* Gas line goes directly to storage or power plant

FIG. 1-5. Coal, oil, and gas must be removed from the ground, cleaned or refined, and then shipped to the point of consumption. These are costly operations.

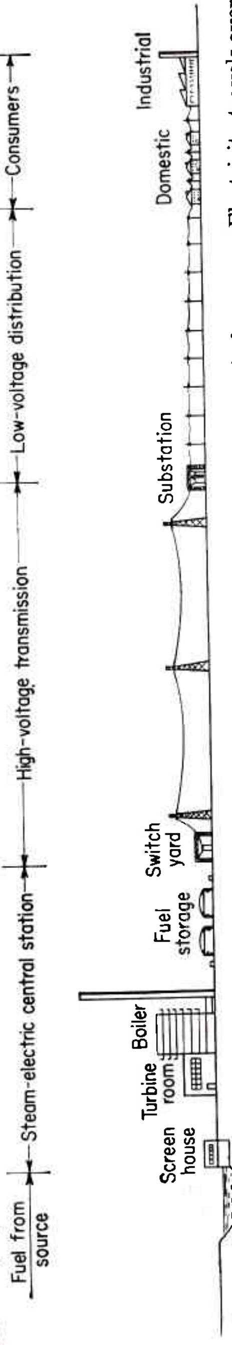


FIG. 1-6. Central power stations receive fuel from mines or refineries and burn it to generate electrical energy. Electricity travels over transmission and distribution systems for delivery to consumers.

atoms and would cost about $140 per pound of heavy hydrogen to produce. This assures an unlimited and very cheap supply of nuclear fuel. The practical problems make this energy source a likely one in only the distant future. To start the reactions in the heavy-hydrogen plasma, it must first be heated to at least 100 million degrees Fahrenheit, and, to produce power, its temperature must rise to about 700 million degrees Fahrenheit. This compares with the boiler-furnace temperatures on the order of 3000 F that we use in our present-day power plants.

COAL

1-4. Coal Fields and Locations. Modern steam boilers burn coal in any of its forms as a primary fuel. Coal developed from vegetable matter which grew in past geological ages. Trees and plants falling into water decayed and produced peat bogs. Gigantic geological upheavals buried these bogs under layers of silt.

Soil pressure, heat, and movement of the earth's crust distilled off some of the bog's gaseous matter to form brown coal, or lignite. Continuing subterranean activity reduces the coal's gaseous content progressively to form different *ranks:* peat, lignite, bituminous, and anthracite. They have formed in many sections of the earth's crust and to a great extent in the United States. Figure 1-7 shows the coal deposits in this country.

1-5. Coal Composition and Analyses. Coal has these components distributed throughout its mass: (1) coal substance, (2) mineral matter, and (3) moisture. The coal substance consists of many organic compounds of carbon, hydrogen, and oxygen derived from the original vegetable matter. The exact composition varies but does not affect practical coal burning. The mineral matter to some extent was in the original plant fibers. Water, which percolated through the peat bogs and coal seams, contained dissolved salts and deposited most of the mineral content of the coal. The moisture content of coals, often called "mechanical moisture," means water retained by the coal. To analyze a coal for these three contents is not useful or feasible. Ultimate and proximate analyses are used instead.

The *ultimate analysis* shows the chemical elements that comprise the coal substance, together with the items ash and moisture. The analysis as made by a chemist shows the following components on a weight-percentage basis with symbols as indicated:

C = carbon
H = hydrogen
O = oxygen
N = nitrogen
S = sulfur
A = ash
W = moisture

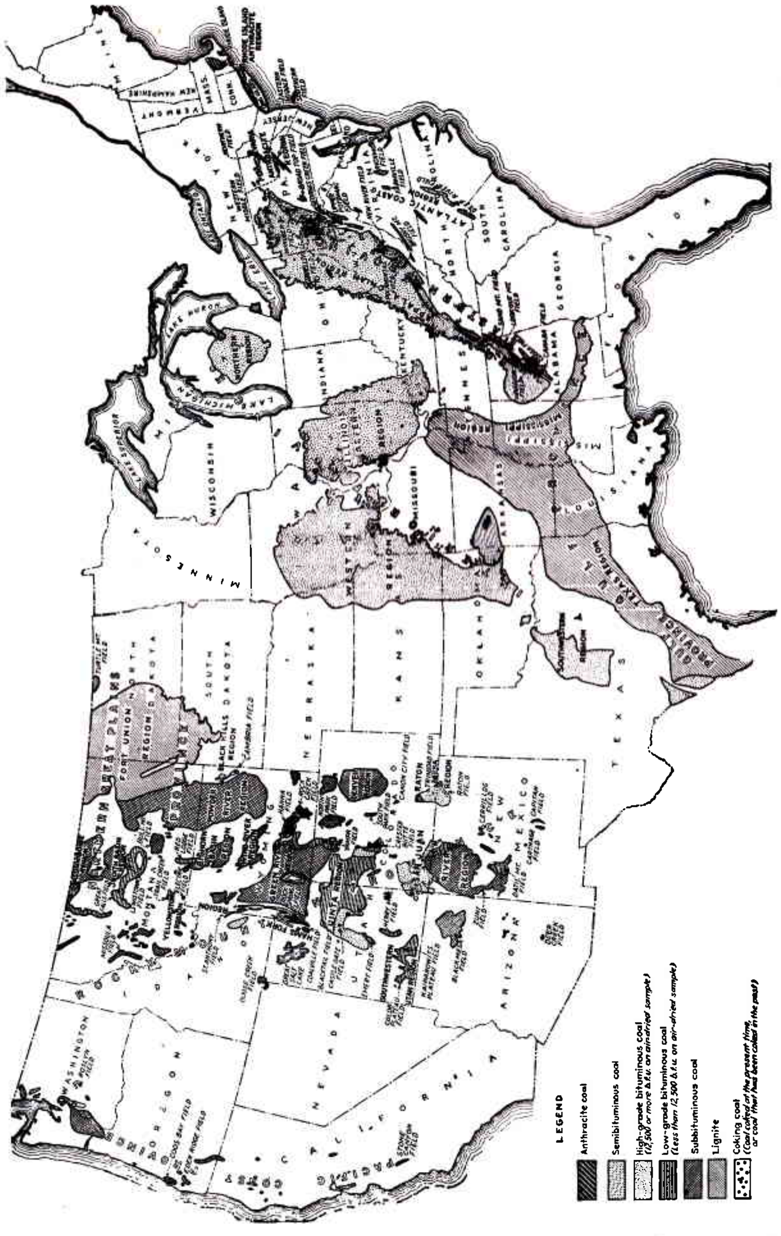

LEGEND
Anthracite coal
Semibituminous coal
High-grade bituminous coal (12,500 or more B.t.u. on air-dried sample)
Low-grade bituminous coal (Less than 12,500 B.t.u. on air-dried sample)
Subbituminous coal
Lignite
Coking coal (Coal coked at the present time, or coal that has been coked in the past)

FIG. 1-7. Various ranks of coal are found in many areas of the United States. (*Courtesy of Keystone Coal Buyers Manual.*)

The moisture W is the liquid mechanically entrained by the coal. The ash A is the substance that remains after coal has been burned completely. It differs somewhat from the weight of the mineral matter, whose water of hydration is lost and whose iron pyrites content oxidizes during burning. The ash item also includes any foreign matter as clay, slate, or dirt that may have fallen into the coal. After burning coal in a furnace the solid refuse consists of this ash plus some unburned combustible material.

In the ultimate analysis the sum of the percentages is 100 since oxygen is found by difference after the other items have been determined. Ultimate analyses may be shown on three different bases—"as-received," "dry-coal," and "combustible." When all the items are shown, it is an ultimate analysis on an as-received or "as-fired" base. An analysis on the dry-coal basis shows the following items as a percentage by weight:

$$C + H + O + N + S + A = 100$$

It is calculated from the as-received values by dividing each of the as-received items except W by the factor $1 - (W/100)$.

Example 1-1. A bituminous coal has the following as-received analysis:

$$\begin{array}{ll} C = 80.5 & S = 1.2 \\ H = 4.1 & A = 5.3 \\ O = 3.0 & W = 4.4 \\ N = 1.5 & \end{array}$$

Calculate the analysis on the dry basis.

Solution. Use the factor

$$1 - \frac{4.4}{100} = 0.956$$

Dry-basis analysis:

$$\begin{aligned} C &= \frac{80.5}{0.956} = 84.2 \\ H &= \frac{4.1}{0.956} = 4.3 \\ O &= \frac{3.0}{0.956} = 3.1 \\ N &= \frac{1.5}{0.956} = 1.6 \\ S &= \frac{1.2}{0.956} = 1.3 \\ A &= \frac{5.3}{0.956} = 5.5 \\ & \qquad\quad \overline{100.0} \end{aligned}$$

The combustible basis shows the following items:

$$C + H + O + N + S = 100$$

The values are obtained by applying the factor $1 - (W + A)/100$ to the as-received analysis. Samples of coal generally lose moisture until an equilibrium is established with the air moisture. If a coal is so affected and then analyzed, we have an "air-dried" analysis which shows all the items, but with values slightly different from the as-received basis. The percentage values for all the different bases always add to 100. The ultimate analysis in its various forms is used for combustion calculations, coal comparisons, and classifications.

The *proximate analysis* of coal shows the following factors on a weight-percentage basis with symbols as indicated:

FC = fixed carbon
VCM = volatile combustible matter
A = ash
W = moisture

This analysis is easily made. The ash and moisture items are the same as in the ultimate analysis and are obtained by completely burning and drying a sample of coal. The volatile combustible matter is found by heating and weighing a sample in a standard test until the volatile gases and moisture are driven off. Fixed carbon is figured by difference, $FC = 100 - W - A - VCM$, and the percentages always add to 100.

The proximate analysis may be given also on the as-received, dry, or combustible basis. Any form of this analysis usually has a separate item showing the percentage of sulfur in the sample. The proximate analysis gives more of a physical picture of the coal and is simple to determine. It is used in some classifications and helps in predicting the behavior of coals during the actual combustion process. Empirical methods for converting a proximate to an ultimate analysis have been established. A knowledge of the source of the coal together with assumptions as to its nitrogen and sulfur content is necessary in addition to the proximate analysis. Very little practical use is made of these conversions. Table 1-1 shows the range of values for ultimate and proximate analyses of some coals.

An analysis of coal, useful in coal classifications, showing composition of the coal substance as distinct from moisture and mineral matter, can be obtained by the Parr formulas. This analysis shows the percentage of fixed carbon and volatile matter in the true coal substance. The sense of fixed carbon and volatile matter is the same as in the proximate analysis, but they relate only to that part of coal containing no moisture or mineral matter. This analysis is on the "moisture- and mineral-matter-free," or "unit-coal," basis. The formulas are

$$\text{Dry, } Mn\text{-free, fixed carbon, }\% = \frac{FC - 0.15S}{100 - (W + 1.08A + 0.55S)} \times 100$$

$$\text{Dry, } Mn\text{-free, volatile matter, }\% = 100 - \text{dry, } Mn\text{-free } FC$$

$$\text{Moist, } Mn\text{-free, heating value, Btu per lb} = \frac{\text{Btu} - 50S}{100 - (1.08A + 0.55S)} \times 100$$

where Mn = mineral matter
FC = as-received fixed carbon, %
W = as-received moisture, %
S = as-received sulfur, %
A = as-received ash, %
Btu = as-received heating value, Btu per lb

A brief extract of some typical ultimate and proximate analyses is given in Table 1-1. These are from the U.S. Bureau of Mines. Note that the hydrogen and oxygen of the ultimate analysis include these elements present in the mechanical moisture.

TABLE 1-1. ANALYSES OF TYPICAL AMERICAN COALS

State	County	Proximate				Ultimate					Btu per lb as-received
		W	VCM	FC	A	C	H	O	N	S	
Anthracite											
Pa.	Schuylkill	2.80	1.16	88.21	7.83	84.36	1.89	4.40	0.63	0.89	13,298
Alaska	Bering River	7.43	6.86	71.35	14.36	70.10	3.71	9.76	1.50	0.57	11,891
Semianthracite											
Pa.	Sullivan	3.38	8.47	76.65	11.50	78.43	3.58	4.86	1.0	0.63	13,156
Va.	Montgomery	4.80	10.12	67.05	18.03	69.27	3.91	7.50	0.66	0.63	11,961
Semibituminous											
W.Va.	Fayette	3.71	23.53	69.37	3.39	82.06	5.14	7.14	1.47	0.80	14,306
Pa.	Somerset	1.10	15.80	75.69	7.41	81.98	4.20	3.56	1.36	1.49	14,499
Colo.	Pitkin	0.96	21.49	68.93	8.62	79.61	4.66	4.76	1.83	0.52	14,330
Bituminous											
Ohio	Jefferson	3.53	37.45	49.90	9.12	71.66	5.15	9.29	1.31	3.47	13,072
Ill.	Saline	7.81	33.54	50.27	8.38	67.40	5.31	15.11	1.44	2.36	12,418
Iowa	Wapello	8.24	30.74	45.02	16.00	59.82	4.81	13.40	0.94	5.03	11,027
Kans.	Atchison	6.95	35.70	45.16	12.19	62.74	5.25	10.74	1.04	8.04	11,905
Ind.	Pike	12.11	34.19	46.87	6.83	65.57	5.70	19.32	1.14	1.44	11,952
N.Mex.	McKinley	12.29	34.58	46.14	6.99	63.31	5.82	22.22	1.03	0.63	11,252
Subbituminous											
Utah	Summit	14.20	36.00	44.80	5.00	61.40	5.79	25.31	1.09	1.41	10,630
Wyo.	Rock Springs	18.86	29.17	47.85	4.12	58.96	5.64	29.34	1.45	0.49	10,283
Lignite											
Tex.	Houston	34.70	32.23	21.87	11.20	39.25	6.93	41.11	0.72	0.79	7,056
N.Dak.	Williams	44.17	23.79	26.28	5.76	36.11	7.28	49.66	0.63	0.56	6,035

1-6. Coal Properties. The moisture, ash, and sulfur contents of coal are undesirable because of their effects on transportation charges, furnace operation, and corrosion of metal surfaces. Coal has other properties that require evaluation. They are (1) *heating value*, (2) *ash-fusion temperature*, and (3) *grindability*. The higher heating value *HHV* is determined either by calorimeter measurements, which give the most accurate values, or by Dulong's equation:

$$HHV = 14{,}544C + 62{,}028\left(H - \frac{O}{8}\right) + 4{,}050S \qquad \text{Btu per lb}$$

where C, H, O, S, are from the ultimate analysis in pounds per pound of coal. The equation gives fairly accurate results for anthracite and the better bituminous coals, but results for low-rank bituminous coals and lignite are in error. Higher heating values of coal range from 11,000 to about 14,000 Btu per lb.

Lower heating values are found by deducting, from the higher heating value, the heat needed to vaporize the mechanical moisture and the moisture formed when a fuel burns. This total moisture equals $W + 9H$.

Ash-fusion temperature is found by a standardized test wherein ash samples are heated in a furnace. Since the ash first softens and then melts, several values of temperature from initial deformation to fluid temperature may be obtained. The value commonly given for a coal refers to the fluid temperature.

Grindability of a coal is shown by a number based upon some arbitrary tests that indicate the ease with which a coal may be pulverized. At present two methods—the Hardgrove and the U.S. Bureau of Mines ball-and-mill method—are approved by the American Society for Testing Materials (ASTM).

Some attempts have been made to evaluate other coal properties significant in the combustion process. These are the "coking," "caking," "free-burning," "friability," and "clinkering" properties.

Coking refers to the ability of the coal, when heated, to evolve volatile matter. A somewhat porous solid of fixed carbon and ash remains, which is called "coke." The formation of coke may be accompanied by the coal's becoming sticky, with pieces adhering to each other. A coal acting in this way is a caking coal. One that burns without this caking tendency is a free-burning coal. A friable coal breaks into smaller pieces when handled. Clinkering refers to a formation of ash which, through softening, fuses together to form a hard and often large lump that clogs combustion equipment. In general, no precise measurements of these properties have been made.

1-7. Coal Classifications. A number of methods for classifying coal have been developed. These methods as related to power plants are, in general, based on rank, size, and source. Figure 1-8 on the combustible basis shows in a general way the progressive change from peat to anthracite and graphite carbon. The bulk of the coal burned in power plants is in the bituminous grouping. Ralston's method of classifying coal is shown in Fig. 1-9. This is a trilinear plot of various ranks of coal on an *A*-, *W*-, *N*-, and *S*-free basis. Any point shows percentages of C, H, and O that add to 100. The plot shows the various ranks of coal within fairly well-defined areas. The most recent method is the ASTM tentative standard shown in Table 1-2, based on physical properties, heating value, and the moisture- and mineral-matter-free basis. Coal can be obtained in various sizes.

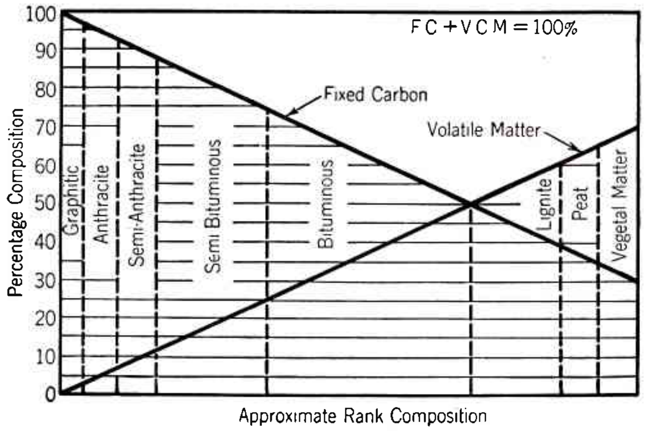


Fig. 1-8. As coal ages, right to left, volatile-matter percentage decreases and fixed-carbon percentage increases.

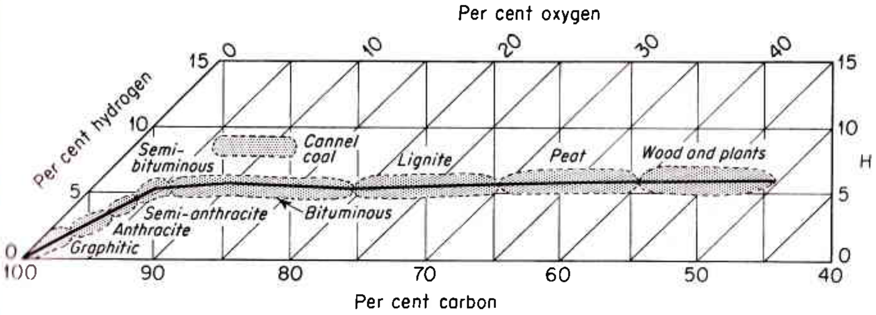


Fig. 1-9. Ralston's chart of coal classification shows that the hydrogen content of coals decreases slowly from the original plant to the bituminous rank, while oxygen decreases rapidly and carbon increases. Beyond the bituminous rank hydrogen decreases as carbon rises.

The United States has deposits of all the different ranks of coal. Figure 1-7 shows their distribution. Most anthracite lies in a limited

TABLE 1-2. CLASSIFICATION OF COALS BY RANK[a]
(ASTM Designation D388-38)

1. **Higher rank coals**, fixed carbon more than 69 per cent, volatile matter less than 31 per cent

Class	Group	Fixed carbon,[b] per cent	Volatile matter,[b] per cent
I. Anthracite	1. Meta-anthracite	98–100	0– 2
	2. Normal anthracite	92– 98	2– 8
	3. Semianthracite	86– 92	8–14[d]
II. Bituminous	1. Low-volatile	78– 86	14–22
	2. Medium-volatile	69– 78	22–31

2. Lower rank coals, fixed carbon less than 69 per cent, volatile matter more than 31 per cent

Class	Group	Limits of Btu[c]
II. Bituminous	3. High-volatile A	14,000 or more
	4. High-volatile B	13,000–14,000
	5. High-volatile C	11,000–13,000[e]
III. Subbituminous	1. Subbituminous A	11,000–13,000[f]
	2. Subbituminous B	9,500–11,000
	3. Subbituminous C	8,300– 9,500
IV. Lignitic	1. Lignite	Less than 8,300[g]
	2. Brown coal	Less than 8,300[h]

[a] This classification does not include a few coals that have unusual physical and chemical properties.

[b] Fixed carbon and volatile matter (dry *Mn*-free basis).

[c] Calorific value on moist *Mn*-free basis.

[d] Nonagglomerating; if agglomerating, classify in low-volatile group of the bituminous class.

[e] There are three varieties of coal in the high-volatile C bituminous coal group: variety 1, agglomerating and nonweathering; variety 2, agglomerating and weathering; variety 3, nonagglomerating and nonweathering.

[f] Both weathering and nonagglomerating.

[g] Consolidated.

[h] Unconsolidated.

region in eastern Pennsylvania. Various bituminous coals and lignite lie to the westward and southward. Coals are often designated by the field, vein, mine, or county from which they come instead of by rank.

From data gathered by exploratory drilling, geologists make estimates as to our coal reserves in the United States. Table 1-3 gives the latest published estimates for the reserves as of Jan. 1, 1953. Total consumption of bituminous and lignitic coals during 1953 in the United States added to 426,798,000 tons. At this rate of consumption our coal resources would last for $(950 \times 10^9)/(0.427 \times 10^9) = 2{,}220$ years.

TABLE 1-3. ESTIMATED COAL RESERVES IN THE UNITED STATES AS OF JAN. 1, 1953

Coal rank	10^9 tons	Corrected to 13,000 Btu per lb, 10^9 tons	Per cent	Estimated heating value, Btu per lb
Anthracite	7	7	1	12,700
Bituminous	525	545	64	13,500
Subbituminous	186	136	15	9,500
Lignite	232	119	20	6,700
Total	950	807	100	

Table 1-3 lists all the coals and assumes they can all be removed. Actually many of the beds and veins are too thin to warrant mining. In many of the mining operations pillars of coal must be left standing to support the roof—so a substantial part of the reserve will not be commercially recoverable. Table 1-4 shows the estimates for all the mineral fuels, on the basis of the commercially recoverable and of the total potential.

TABLE 1-4. ESTIMATED RECOVERABLE MINERAL FUELS OF THE UNITED STATES AS OF JAN. 1, 1953

Fuel	Reserves, 10^{15} Btu		Per cent of total	
	Proved and commercially recoverable	Total potential	Proved and commercially recoverable	Total potential
Coal	5,117	20,469	90.3	84.0
Petroleum and natural-gas liquids	198	423	3.5	1.7
Natural gas	200	487	3.5	2.0
Bitumen from bituminous sandstone	4	8		
Oil from oil shale	150	3,000	2.7	12.3
Total	5,669	24,387	100.0	100.0

Coal by far exceeds the other mineral fuels in our resources. But Fig. 1-1 shows that we use more oil and gas, about 26 quadrillion Btu in 1953. On this basis these fuel resources would last $548/26$, or 21, years. This short period need not alarm us—yet—because oil discoveries keep pace with the rising consumption. Since 1910 we have just about always had a 15- to 25-year reserve in sight. In addition techniques are being developed to convert coal to oil. When petroleum runs short, its price will rise and so make the coal-oil conversion economical.

Rough estimates of uranium resources show that this nuclear fuel will probably represent about 95 per cent of *all* our mineral-fuel energy. Much development is needed to make uranium fuel economical.

OIL

1-8. Oil Fields and Locations. Figure 1-1 shows the major importance of oil and gas as a fuel source for the United States. Oil is the larger of the two. Refining petroleum or crude oil produces our fuel oils. Petroleum comes from many areas in the United States.

Several theories have been offered to explain the origin of petroleum; the most plausible assumes that it is decayed marine life, both vegetable and animal. Several times during past geological ages the sea has partly covered our present continent. This allowed accumulating dead marine carcasses and vegetable matter for millions of years. Between floods sediments covered the deposits. Pressure and heat caused by movement of the earth's crust transformed these deposits into oil.

Oil deposits accumulate in voids in the rocks and sands below the earth's surface. Oil generally has an underlying body of water and pockets of pressurized natural gas above. Fairly dense earth strata cover most deposits. Oil wells drilled through this layer penetrate the deposits. The pressure forces the gas and oil to the surface. After the pressure has diminished, the oil must be pumped.

Figure 1-10 shows the oil fields in the United States. Petroleum may be a paraffinic type, an asphaltic type, or a mixed-base crude. The designation depends on whether a paraffin wax, asphalt, or a mixture of the two remains after distillation. Paraffin-base oils lie in the eastern part of the Eastern district, with some in the Rocky Mountain district. Asphalt-base oils are found in the other regions.

1-9. Oil Composition and Analysis. Petroleum is a mixture of many organic compounds, principally of carbon, hydrogen, and oxygen. No attempt is made to analyze the oil for these compounds for power-plant work; instead an ultimate analysis is used. This shows the percentage by weight of the elements present:

C = carbon
N = nitrogen
H = hydrogen
S = sulfur
O = oxygen

Regardless of the crude-oil source, its composition is fairly uniform within

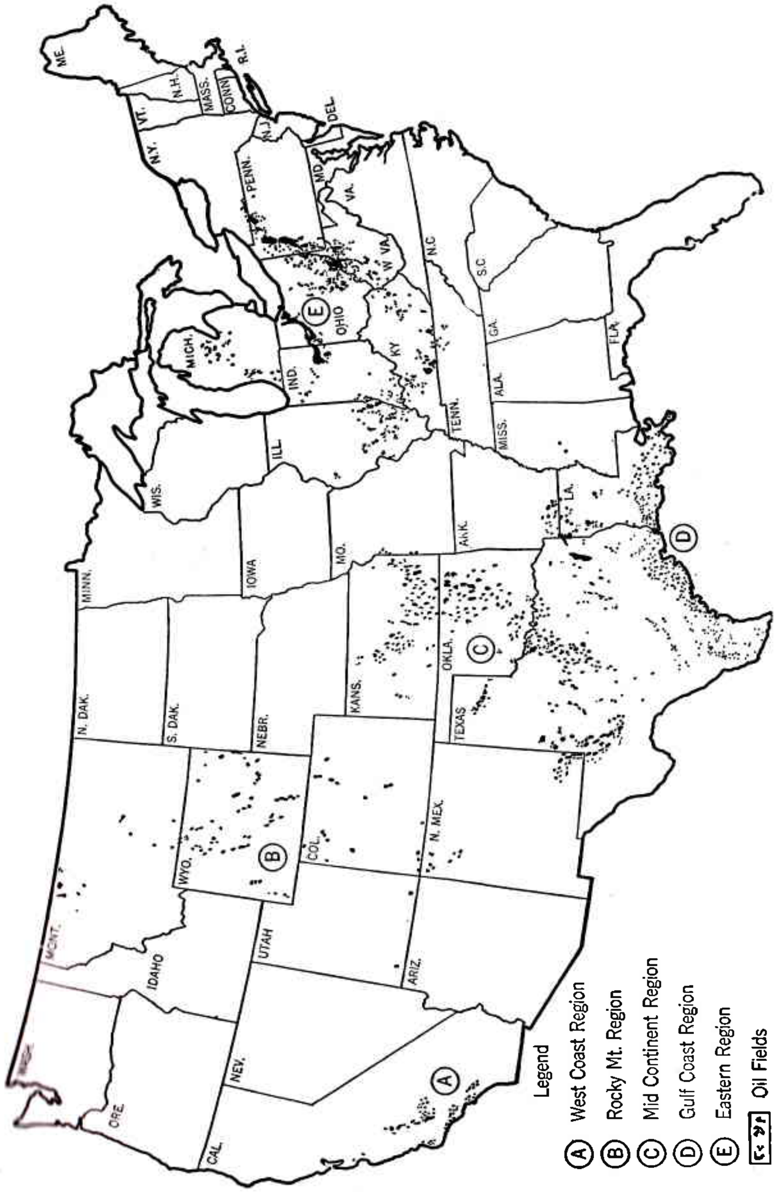


Fig. 1-10. Oil fields concentrate in certain areas of the United States.

close limits. Approximate values are

$$C = 83\text{–}87\% \qquad O + N = 0\text{–}7\%$$
$$H = 11\text{–}16\% \qquad S = 0\text{–}4\%$$

Moisture and sediment contaminate all oils to some extent. These items may be given in the analysis.

Crude oil, or petroleum, is seldom used as such. A refining process produces gasoline, kerosene, gas oil, lubricating distillate, and a residual oil, tar, or petroleum coke. These in turn are processed to produce the wide array of petroleum products found on the market. The products in general are motor fuels, kerosene, fuel oils, lubricating oils, paraffin, asphalt, and coke. Oils used for fuels are of the following types:

1. *Topped crudes.* Petroleum which through weathering has lost some of the lighter oils, or oil from which the gasoline, kerosene, and gas-oil fractions have been removed.
2. *Distillate fuel oils.* Oil distilled off between the kerosene and lubricating-oil fraction.
3. *Residual fuel oils.* Oil left over after most of the lighter fractions have been removed.
4. *Blended fuels.* Mixture of any or all of the foregoing fuels.
5. *Refinery wastes.* Sludges or tank cleanings resulting from the distillation process.

1-10. Oil Properties. Petroleum or any of its derivatives is usually analyzed on the basis of physical rather than chemical properties. These are:

1. *Specific gravity.* Expressed in terms of an arbitrary scale in degrees API (American Petroleum Institute) or degrees Baumé. These are uniform scales on hydrometer floats used for their determination. Relations between scales and specific gravities are:

$$°API = \frac{141.5}{sp\ gr\ (60\ F/60\ F)} - 131.5$$

$$°Bé = \frac{140}{sp\ gr\ (60\ F/60\ F)} - 130$$

For many practical purposes the two scales give essentially the same numerical values. A quoted API or Baumé degree of an oil implies a temperature of 60 F. Values determined at other temperatures are adjusted to 60 F by correction tables such as the National Standard Petroleum Oil Tables.*

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* *National Bureau of Standards Circular* C154.

2. *Heating value.* The higher heating value ranges from about 18,000 to 19,500 Btu per lb. Empirical equations giving approximate values are used. The following applies to heavy cracked fuel oils:

$$HHV = 17{,}645 + 54 \times °API \qquad \text{Btu per lb}$$

3. *Flash point and fire point.* The temperature at which oil vapor flashes or burns steadily; used in relation to ignition and storage hazards of oil.
4. *Pour point.* The lowest temperature at which oil flows; of use in pumping and flow of oils.
5. *Viscosity.* Useful in any of several standard units in connection with atomization and pumping of oil.
6. *Sulfur content.* In weight percentage; useful with reference to corrosion of plant equipment.
7. *Moisture and sediment.* Weight or volume percentage; useful in the firing of oils.
8. *Specific heat.* Varies with temperature but for usual values is about 0.4 to 0.5 Btu per lb-°F; used in oil heating problems.
9. *Coefficient of voluminal expansion.* Varies with temperature, but usually about 0.0004 per degree Fahrenheit;* used in oil purchasing and storage, since oil is bought by volume usually expressed in terms of 42-gal barrels. The equation used is

$$V_t = V_0[1 + 0.0004(t - t_0)]$$

where V_t = volume at temperature t, °F
V_0 = volume at temperature t_0, °F

10. *Carbon residue.* Test indicating carbon-forming characteristic of an oil; used in oil burning.

The foregoing properties are all determined from standardized laboratory tests given in references. These references must also be used if exact values of properties are needed for a given set of conditions rather than the average values shown above.

1-11. Oil Classifications. Fuel oils are specified in terms of the U.S. Bureau of Standards grades 1 to 6, where No. 6 is the usual power-plant fuel. Each grade has certain ranges of values of some of the various properties listed previously. They are given in the extended classification of the Bureau. Other specifications are sometimes used, among them being the bunker fuel oils, grades A, B, and C, for the United States government. Table 1-5 shows the ASTM specifications for the six grades of fuel oils.

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* See ASTM Designation D206 for precise correction factors.

TABLE 1-5. DETAILED REQUIREMENTS FOR FUEL OILS[a]

Grade of fuel oil[b]	Flash point, F	Pour point, F	Water and sediment, by vol.	Carbon residue on 10% residuum, %	Ash, % by wt.	Distillation temperatures, F: 10% point	Distillation temperatures, F: 90% point	Distillation temperatures, F: End point	Saybolt viscosity, sec: Universal at 100 F		Saybolt viscosity, sec: Furol at 122 F		Kinematic viscosity, centistokes: At 100 F		Kinematic viscosity, centistokes: At 122 F	
	Min	Max	Max	Max	Max	Max	Max	Max	Max	Min	Max	Min	Max	Min	Max	Min
No. 1. A distillate oil intended for vaporizing pot-type burners and other burners requiring this grade[c] API gravity 35 (min)	100 or legal	0	Trace	0.15	....	420	...	625	...	...	...	..	2.2	1.4		
No. 2. A distillate oil for general-purpose domestic heating for use in burners not requiring No. 1 API gravity 26 (min)	100 or legal	20[d]	0.10	0.35	....	[e]	675	...	40	...	...	..	(4.3)			
No. 4. An oil for burner installations not equipped with preheating facilities	130 or legal	20	0.50	....	0.10	...	...	...	125	45	...	..	(26.4)	5.8		
No. 5. A residual-type oil for burner installations equipped with preheating facilities	130 or legal	...	1.00	....	0.10	...	...	...	...	150	40	..	.....	32.1	(81)	
No. 6. An oil for use in burners equipped with preheaters permitting a high-viscosity fuel	150	...	2.00[f]	....	....	...	...	...	...	...	300	45	.....	....	(638)	(92)

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[a] Recognizing the necessity for low-sulfur fuel oils used in connection with heat-treatment, nonferrous-metal, glass, and ceramic furnaces, and other special uses, a sulfur requirement may be specified in accordance with the following table:

Grade of fuel oil	*Sulfur, max, %*
No. 1	0.5
No. 2	1.0
Nos. 4, 5, and 6	No limit

Other sulfur limits may be specified only by mutual agreement between the buyer and seller.

[b] It is the intent of these classifications that failure to meet any requirement of a given grade does not automatically place an oil in the next lower grade unless in fact it meets all requirements of the lower grade.

[c] No. 1 oil shall pass test for corrosion made in accordance with paragraph 15 of ASTM Tentative Specifications for Fuel Oils, D396-48T.

[d] Lower or higher pour points may be specified whenever required by conditions of storage or use. However, these specifications shall not require a pour point lower than 0 F under any condition.

[e] The 10 per cent point may be specified at 440 F maximum for use in other than atomizing burners.

[f] The amount of water by distillation, plus the sediment by extraction, shall not exceed 2.00 per cent. The amount of sediment by extraction shall not exceed 0.50 per cent. A deduction in quantity shall be made for all water and sediment in excess of 1.0 per cent.

NATURAL GAS

1-12. Gas Fields and Locations. Natural gas has a geological history similar to that of petroleum. About one-third of the natural gas produced today is found with petroleum. The pressurized gas forces the oil to the surface when a well taps the strata.

Figure 1-11 shows the main gas fields in the United States. Natural gas proves an economical fuel for many areas. Long-distance pipe lines carry it for distances as far as from Texas to the New England states. Compressor stations along the pipe lines move it, as needed, to the market areas.

About 15 per cent of natural gas is used domestically, another 6 per cent does heating in commercial enterprises, and only about 10 per cent fires central-station boilers. The remainder is used industrially for making carbon black, refractories, glass, and cement and in metal refining and oil refining. Natural gas outranks oil in producing electrical energy in central stations.

1-13. Gas Analysis and Properties. Natural gas is a mixture of several gases; methane (CH_4) and ethane (C_2H_6) are the main constituents. Gases are analyzed on a volumetric basis.

TABLE 1-6. SAMPLES OF NATURAL-GAS ANALYSES

Area	Gas constituent, per cent by volume: CO_2	N_2	CH_4	C_2H_6	Sp gr (ref. to air)	Heating value, Btu per cu ft
Barren County, Ky.	3.8	2.9	23.6	69.7	1.21	1548
Los Angeles, Calif.	25.0	1.4	59.1	14.5	0.88	843
Los Angeles, Calif.	6.5	...	77.5	16.0	0.70	1073
Kiefer, Okla.	2.4	1.8	64.1	31.7	0.97	1272
Abilene, Tex.	0.0	4.6	74.5	20.9	0.72	1129
Kansas City, Mo.	0.8	8.4	84.1	6.7	0.63	967
Moab, Utah	3.6	5.6	90.8	0.0	0.61	967
Caddo Parish, La.	0.9	1.5	97.6	0.0	0.75	1039

Higher heating values vary from about 700 to 1000 Btu per cu ft at standard conditions, usually atmospheric pressure and 60 F. The heating value of any gas can be calculated by adding the product of each component volume (as a decimal) and its individual volumetric heating value.

Table 1-6 shows some gas analyses as distributed in various areas. Gas analyses from one field do not remain constant but vary slowly with time.

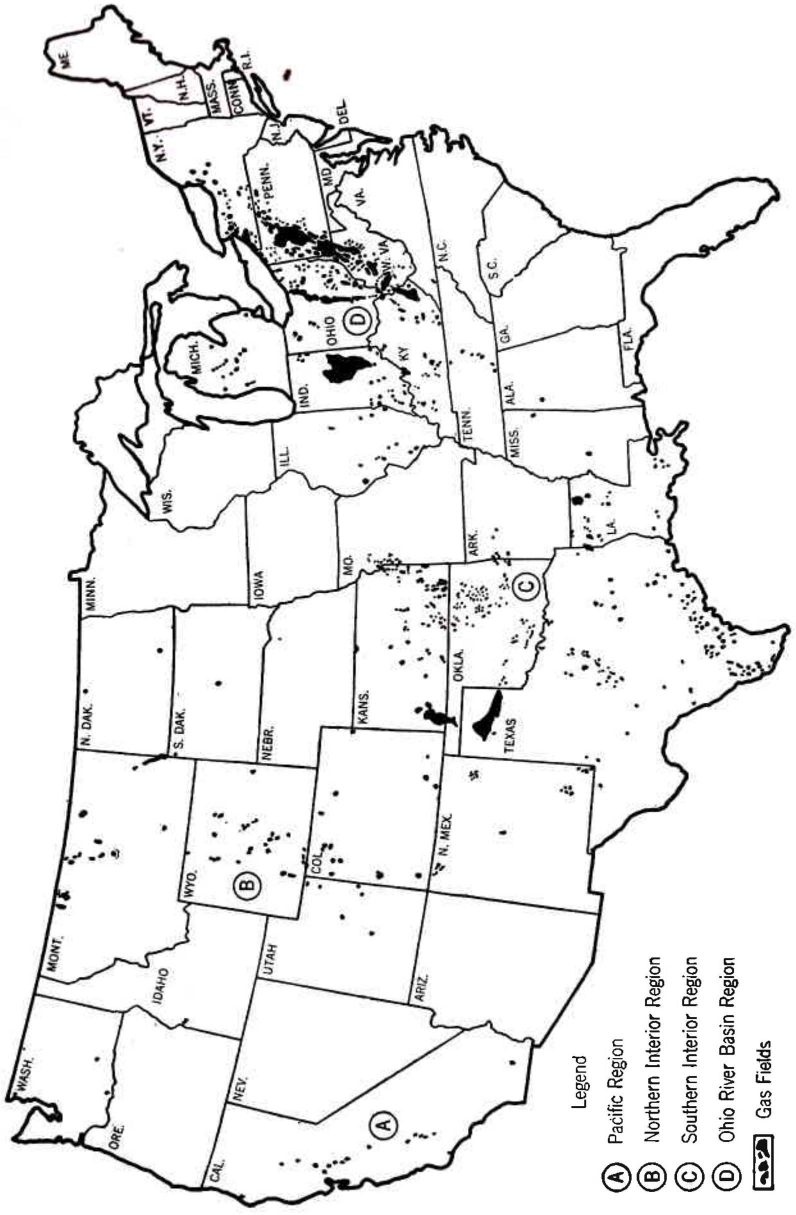

FIG. 1-11. Natural gas is found in the same general areas as petroleum.

MANUFACTURED, BY-PRODUCT, AND REFUSE FUELS

1-14. Types and Sources. Although most of the steam produced is from coal, oil, and gas, there are instances where some other fuel is used. For the generation of *small* quantities of steam, a manufactured fuel such as producer gas, water gas, illuminating gas, or briquettes may be used. When comparatively large quantities of steam are produced from other fuels, they are usually a by-product of some industrial process or a refuse that would normally be discarded. These fuels are used in the power plant of the industries when it is economically feasible to do so. Analyses of these fuels vary widely and are on either the weight or the volume basis depending on whether they are solids, liquids, or gases. Burning these fuels is a distinct problem in the design of combustion equipment, furnaces, and to some extent heat-transfer surfaces. The moisture contents, contaminations, corrosion from sulfur gases, and low heating values in many cases are elements that tend to make the combustion equipment differ somewhat from the conventional type. Table 1-7 lists some of these fuels; heating values must usually be determined by test.

TABLE 1-7. MANUFACTURED, BY-PRODUCT, AND REFUSE FUELS

Solids	*Liquids*	*Gases*
Wood waste	Colloidal fuel	Producer gas
Coke	Sulfite liquor	Water gas
Coke breeze	Acid sludge	Blast-furnace gas
Oil shale	Tar	Illuminating gas
Tanbark	Tank cleanings	Coke-oven gas
Bagasse	Alcohol	Sewage gas
Rice hulls	Shale oil	
Straw		
Charcoal		
Briquettes		
City refuse		
Petroleum coke		
Bark		
Dry sewage sludge		

FUEL PURCHASING

1-15. Fuel Selection. Steam generators and internal-combustion (i-c) engines are designed to burn specific types of fuel. Heating value, sulfur content, ash-fusion temperatures, ash and moisture content, and other factors varying from the specified amounts may seriously limit the capacity of the equipment and even make it inoperative.

Many modern boilers can burn coal, oil, and gas either separately or simultaneously. Similarly some i-c engines can burn distillate oil or natural gas or both. This allows taking advantage of price fluctuations for maximum operating economy.

Factors to be considered in buying fuels include: (1) energy content, (2) price, (3) unit operating efficiency, (4) handling cost, (5) refuse disposal, (6) operating labor, and (7) maintenance.

1. Though fuels are purchased in terms of tons or barrels, the significant factor is their total energy content. This may be calculated by multiplying their weight by the unit heating value.

2. Fuel prices are quoted at so many dollars per ton, barrel, or 1,000 cu ft. These prices are not commensurate, and so they must be converted to the price per unit of energy. The most common method reduces all fuel prices to cents per million Btu; for example, for coal:

$$\text{Cents per million Btu} = \frac{\text{dollars per long ton} \times 100 \times 10^6}{2{,}240 \times HHV}$$

where HHV is expressed in Btu per pound.

Plants using large quantities of fuel generally buy directly from the mining or producing company. The price quoted is generally f.o.b. at the source or fuel-preparation plant. The ultimate cost to the consumer of any fuel may be several times the cost at the source, the difference being the cost of handling and transportation from the source to the point of consumption via rail, truck, boat, pipe line, or any combination of these. Electric central stations located at the mine mouth eliminate intermediate handling costs. But this has been practical in few instances. In most cases the cost of providing cooling water at the mine or of electric-transmission facilities to the load center proves greater than the cost of coal transportation to a plant near the load center.

3. The operating efficiency of a heat cycle or engine inherently depends upon three main factors: its design, the nature of the fuel, and the skill of the operator. In selecting a fuel considerable care and testing are required to choose the most economical one. For example, coal properties that will require scrutiny are the proportions of volatile matter, ash, sulfur, and moisture; ash-fusion temperature; ash analysis; size of lumps; friability; and grinding and coking characteristics. All these have a bearing on the efficiency with which a boiler will perform with a given fuel. In addition they control other operating features, such as maximum steaming capacity, slagging, clinkering, smoke, draft requirements, corrosion, and dust nuisance. Aside from their effect on operating efficiency these factors may involve added expense for dust, smoke, and handling equipment. Many fuels will be unsuitable owing to some one or more properties unduly deviating from required limits. A tentative selection can usually be reduced to three or four fuels, trial shipments of which will then be tested in the plant for efficiency and operating flexibility.

Not only gross boiler efficiency must be considered, but the effect on over-all plant efficiency. For example, if the caking characteristics of a coal melt the mixture of coke and ash and mat the fuel bed, it will require additional fan power to force the combustion air through the partly fused bed of coals. This reduces plant efficiency by requiring a greater amount of power for fan drive and also reduces the maximum capacity.

4. General experience shows that, within the plant, gas entails the least handling cost, oil is intermediate, and coal requires the most.

Gas supply requires little more than a connection to the mains, meters, reducing valve, and stop or control valves before it is ready for the burners. Oil requires a storage tank, pumps for unloading and transfer to the burners, piping and valves, and usually additional heating equipment to make the oil less viscous to increase ease of handling. Coal requires unloading equipment, bunkers, weigh scales, and distributing lorries or belt-conveyer systems. For pulverized coal, grinding mills must be provided.

5. In general oil and gas fuels do not present any refuse-disposal problem since their ash content is negligible and passes out with the flue gases. For coal, ashes must be moved from the furnaces to some central storage location within the plant. Suitable loading equipment for disposal into cars or boats must be provided. At times ash and cinders are in demand for building purposes and may be sold for a cash return.

6. The labor required by each type of fuel will depend on the handling and cleaning needs. Gas requires the least in additional operating labor and coal the most, oil being intermediate. For coal-fired plants the coal with the greatest ash content and the lowest fusion temperature will cause the greatest fouling of boiler heating surfaces and the most clinker formations in fuel beds.

7. The effect of different fuels on maintenance is the most difficult factor to evaluate, as only extended experience with a given fuel can give the answer. The maintenance needed depends on the operating policy and the skill of the operators, as well as the type of fuel burned. Experience at each plant is usually unique and cannot confidently be used to evaluate the effect of fuel at another plant.

1-16. Sampling. Fuels in general are not homogeneous substances. Oil from a given well or coal from a given mine will vary in the different properties from batch to batch, coal more so than oil. As found in the earth, coal seams are made up of strata, termed "benches," lying one above the other and often separated by relatively thin strata of slate, rock, or bone coal (coal with an extremely high ash content). The analysis of coal from various benches at the same seam face, or cross section, may show markedly different proportions of volatile matter, fixed carbon, sulfur, and ash content and, as a result, variations in heating value. Before the introduction of mechanical coal loading in the mines at the seam face the coal from each bench was readily segregated by manual sorting, and the coal sent to the consumer was fairly uniform in analysis. With increasing mechanization of mining, coal varies more in its characteristics, owing to more mixing of coal from various benches and less discriminating selection at the seam face.

Finding the characteristics of a shipment of fuel is a problem in itself. The solution lies in choosing methods that will yield a sample representative of the whole shipment. The sampling methods used are codified by the ASTM in their standards. A truly representative sample must contain, in the same proportion, all the elements in the original batch of fuel. The larger the sample, the smaller the chance of error in determining the true properties of the given batch of fuel. However, the laboratory methods used for analysis and the economic need of reducing the amount of fuel diverted from its ultimate use dictate that the sample shall be small. Error due to sampling can be reduced to a few tenths of a per cent by the following method: For coal, depending upon the size of the lumps, a gross sample is first taken during unloading. For example, a sample is taken by removing a scoopful for every 5 tons. This gross sample, which may weigh a ton for a shipment of several hundred tons, is then thoroughly mixed, the larger lumps being first crushed to a certain minimum size. The gross sample is then reduced by discarding alternate shovelfuls. The remaining pile is again thoroughly mixed and further crushed. This sample is reduced by repeating the operation of discarding alternate shovelfuls. Successive reduction of the sample finally brings it down to about 5 lb in a pulverized form, when it is ready for laboratory analysis. Motor-driven sample crushers do all these operations in one continuous process.

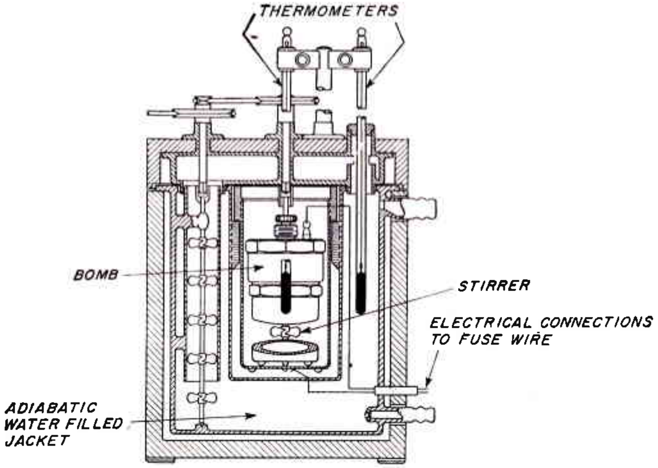
THERMOMETERS
BOMB
STIRRER
ELECTRICAL CONNECTIONS TO FUSE WIRE
ADIABATIC WATER FILLED JACKET

FIG. 1-12. Weighed fuel sample burning inside bomb heats water of known amount surrounding bomb. Temperature rise of water measures amount of heat.

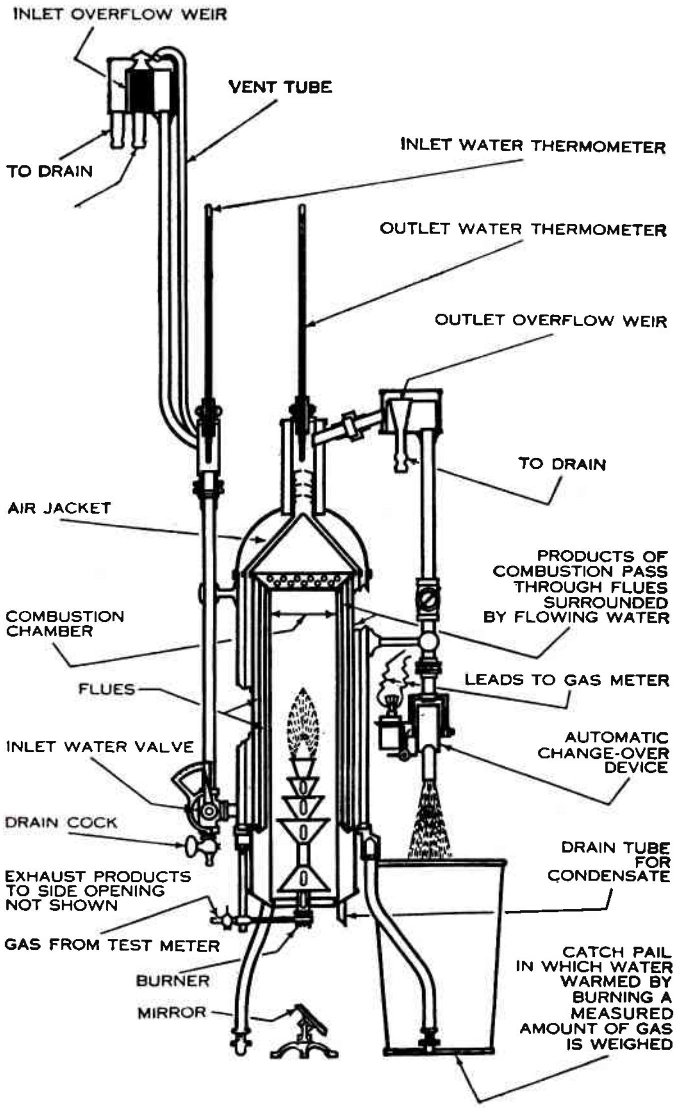
INLET OVERFLOW WEIR
VENT TUBE
TO DRAIN
INLET WATER THERMOMETER
OUTLET WATER THERMOMETER
OUTLET OVERFLOW WEIR
TO DRAIN
AIR JACKET
PRODUCTS OF COMBUSTION PASS THROUGH FLUES SURROUNDED BY FLOWING WATER
COMBUSTION CHAMBER
LEADS TO GAS METER
FLUES
AUTOMATIC CHANGE-OVER DEVICE
INLET WATER VALVE
DRAIN COCK
DRAIN TUBE FOR CONDENSATE
EXHAUST PRODUCTS TO SIDE OPENING NOT SHOWN
GAS FROM TEST METER
CATCH PAIL IN WHICH WATER WARMED BY BURNING A MEASURED AMOUNT OF GAS IS WEIGHED
BURNER
MIRROR

FIG. 1-13. Measured flow of oil or gas heats up a known flow rate of water. Water-temperature rise measures amount of heat developed. (*Courtesy of Precision Scientific Company.*)

Contracts for purchase of large amounts of coal include specifications of the coal characteristics and usually a penalty clause if the coal does not conform to stated limits. It is usual to allow a difference of 2 per cent in the weights of the coal as measured at the mine and as measured at point of delivery. This tolerance allows for loss or gain in moisture during transportation.

1-17. Calorimetry. The heating value of any fuel is the energy given up during the *complete* combustion of 1 lb of the fuel and the cooling of the combustion products back to the ambient fuel temperature before firing. A calorimeter measures the heating value of a fuel by absorbing in a known amount of water all the heat given up by the burning of a weighed fuel sample.

Multiplying the water weight by the temperature rise evaluates the heat absorbed, which equals the heat of combustion yielded by the burned fuel sample. Two principal types of calorimeters are used commercially: (1) The constant-volume nonflow type and (2) the constant-pressure steady-flow type. The first is the *bomb-type* calorimeter for measuring coal and oil heating values. The second is the Junker calorimeter used for fluid fuels, oil and gas.

In the bomb calorimeter (Fig. 1-12) about 1 g of the fuel being tested burns in a bomb that can withstand internal pressures in excess of 1,000 psi. The bomb may be about 6 in. in its longest dimension. The bomb comes apart and has an internal rack to receive a small tray carrying the weighed fuel specimen. Oxygen forced into the assembled bomb raises pressure up to about 500 psi. After placing the bomb in a measured amount of water (about 2,000 g) a spark initiates combustion. The fuel's heat transfers through the bomb walls into the water. From the temperature rise and weight of the water and the weight of the fuel, and by making allowances for radiation and other factors, the fuel heating value can be calculated accurately.

In the Junker calorimeter (Fig. 1-13) the oil or gas burns in a bunsen-type burner under atmospheric conditions. The combustion products flow through passages surrounded by water until their temperature falls to that of the surrounding air. The amount of fuel burned is metered for a specific length of time. The water absorbing the heat of combustion flows continuously and is weighed during the duration of the test. From the metered water and fuel and observed temperature rise the heating value may be accurately calculated.

CHAPTER 2

COMBUSTION

2-1. Combustion Chemistry. The combustible elements in fuels consist of carbon and hydrogen with a small amount of sulfur. Fuels may be solid, liquid, or gaseous before we burn them. Combustion involves a complicated series of chemical reactions forming intermediate products and ending in relatively simple combustion gases. For power purposes we regard combustion as the rapid high-temperature oxidation of the burnable elements that release heat energy.

Oxygen needed for the oxidation processes can be calculated as follows:

$$\begin{aligned} C + O_2 &= CO_2 \\ 12 \text{ lb} + 32 \text{ lb} &= 44 \text{ lb} \\ 1 \text{ lb} + 2.67 \text{ lb} &= 3.67 \text{ lb} \\ 1 \text{ mole } C + 1 \text{ mole } O_2 &= 1 \text{ mole } CO_2 \\ 1 \text{ vol.} + 1 \text{ vol.} &= 1 \text{ vol.} \end{aligned} \tag{2-1a}$$

$$\begin{aligned} 2H_2 + O_2 &= 2H_2O \\ 4 \text{ lb} + 32 \text{ lb} &= 36 \text{ lb} \\ 1 \text{ lb} + 8 \text{ lb} &= 9 \text{ lb} \\ 2 \text{ moles } H_2 + 1 \text{ mole } O_2 &= 2 \text{ moles } H_2O \\ 2 \text{ vol.} + 1 \text{ vol:} &= 2 \text{ vol.} \end{aligned} \tag{2-1b}$$

$$\begin{aligned} 2C + O_2 &= 2CO \\ 24 \text{ lb} + 32 \text{ lb} &= 56 \text{ lb} \\ 1 \text{ lb} + 1.33 \text{ lb} &= 2.33 \text{ lb} \\ 2 \text{ moles } C + 1 \text{ mole } O_2 &= 2 \text{ moles } CO \\ 2 \text{ vol.} + 1 \text{ vol.} &= 2 \text{ vol.} \end{aligned} \tag{2-1c}$$

$$\begin{aligned} S + O_2 &= SO_2 \\ 32 \text{ lb} + 32 \text{ lb} &= 64 \text{ lb} \\ 1 \text{ lb} + 1 \text{ lb} &= 2 \text{ lb} \\ 1 \text{ mole } S + 1 \text{ mole } O_2 &= 1 \text{ mole } SO_2 \\ 1 \text{ vol.} + 1 \text{ vol.} &= 1 \text{ vol.} \end{aligned} \tag{2-1d}$$

The values given below each numbered equation show the relative parts by weights, moles, and volumes of the constituents taking part in the reaction. The volumes assume all constituents to be gaseous. A mole refers to a pound mole based on the molecular weight of a substance.

Mass conservation requires the total *weights* on each side of an equation to be equal. The number of *moles*, however, need not be equal. One mole of a gas or gas mixture occupies 359 cu ft at 32 F and 14.7 psia. Significantly, the ratio of the number of moles of the different substances is also the ratio of their volumes. Hence the per cent by volume of gas in a mixture equals the ratio of the moles of that gas to the total moles of the mixture.

Reaction equations for the oxidation of compounds can also be written.

$$\begin{aligned} 2C_2H_6 + 7O_2 &= 4CO_2 + 6H_2O \\ 60\text{ lb} + 224\text{ lb} &= 176\text{ lb} + 108\text{ lb} \\ 2\text{ moles} + 7\text{ moles} &= 4\text{ moles} + 6\text{ moles} \end{aligned}$$

A hypothetical mole is also used for solid or liquid fuels if their weight analysis by elements is known.

Example 2-1. A fuel oil has the following weight-percentage analysis:

$$\begin{matrix} C = 83.0 & O = 3.0 & S = 1.0 \\ H = 11.0 & N = 2.0 & \end{matrix}$$

Determine the mole formula, and write the complete reaction equation with oxygen.

Solution. For convenience let us deal with 100 lb of fuel. The moles of each element in the fuel oil will be inversely proportional to its molecular weight; then:

Element	Weight	Molecular weight	Moles
C	83	12	6.92
H	11	2	5.5
S	1	32	0.031
O	3	32	0.094
N	2	28	0.071
Total	100	. . .	12.616

To burn 100 lb of fuel oil, we must satisfy

$$12.616(C,H,S,O,N) + M_0O_2 = 6.92CO_2 + 5.5H_2O + 0.031SO_2 + 0.071N_2$$

The oxygen weight must be the same on each side of the equation; so

$$\begin{aligned} 0.094 \times 32 + M_0 \times 32 &= 6.92 \times 32 + 5.5 \times 16 + 0.031 \times 32 + 0.071 \times 0 \\ M_0 &= 9.61\text{ moles }O_2 \end{aligned}$$

If the composition of a fuel is known, Eqs. (2-1) can be used to calculate oxygen needs. This can be reduced to the theoretical (stoichiometric) oxygen requirement,

$$W_0 = 2.67C + 8\left(H - \frac{O}{8}\right) + S \qquad \text{lb per lb fuel} \qquad (2\text{-}2)$$

The symbols C, H, O, S are weight fractions of the elements as given by an ultimate fuel analysis. For gas fuels the volumetric analysis must be converted to a weight analysis as follows:

$$C = \frac{12CH_4 + 12CO + 24C_2H_4 + \cdot \cdot \cdot}{MW} \qquad (2\text{-}3a)$$

$$H = \frac{2H_2 + 4CH_4 + \cdot \cdot \cdot}{MW} \qquad (2\text{-}3b)$$

$$O = \frac{16CO + 32CO_2 + \cdot \cdot \cdot}{MW} \qquad (2\text{-}3c)$$

$$S = \frac{32H_2S + 32SO_2 + \cdot \cdot \cdot}{MW} \qquad (2\text{-}3d)$$

where the symbols CH_4, CO, H_2, H_2S, etc., are volume fractions of the compound constituents and MW is the molecular weight of the gas mixture, or

$$MW = 16CH_4 + 28CO + 2H_2 + 28C_2H_4 + 44CO_2 + 34H_2S + \cdot \cdot \cdot$$

to include all components. Constants are listed in Table 2-1.

The pound-*weight* and pound-*mole* basis for computing oxygen needs are shown below.

Example 2-2. Determine the theoretical oxygen needed for the complete combustion of 1 lb of coal of the following ultimate analysis:

$$\begin{matrix} C = 71.7 & O = 8.3 & S = 3.4 & W = 3.5 \\ H = 5.2 & N = 1.3 & A = 6.6 & \end{matrix}$$

Solution. By Eq. (2-2)

$$\begin{aligned} W_0 &= 2.67 \times 0.717 + 8\left(0.052 - \frac{0.083}{8}\right) + 1 \times 0.034 \\ &= 2.28\text{ lb} \end{aligned}$$

By Eqs. (2-1)

Elements	O_2, lb per lb fuel	O_2, moles per lb fuel
C	$2.67 \times 0.717 = 1.914$	$1 \times 0.717/12 = 0.0598$
H	$8 \times 0.052 = 0.416$	$0.5 \times 0.026 = 0.0130$
S	$1 \times 0.034 = 0.034$	$1 \times 0.034/32 = 0.0011$
Total	2.364 lb	0.0739 mole

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Oxygen required from weight analysis:

$$O_2 = 2.364 - 0.083^* = 2.28 \text{ lb}$$

Oxygen required from mole analysis:

$$O_2 = 0.0739 - \frac{0.083^*}{32} = 0.0713 \text{ mole per lb fuel}$$
$$= 0.0713 \times 32 = 2.28 \text{ lb}$$

2-2. Air for Combustion. Oxygen for combustion in power furnaces comes from the atmospheric air. In furnace design we must calculate the air flow needed to support combustion. Figure 2-1 outlines the combustion process using air. Air data we must know include:

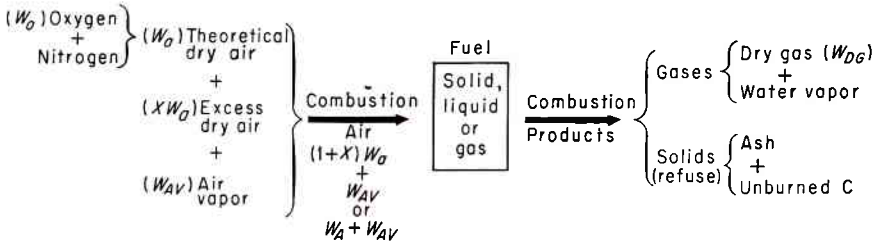

FIG. 2-1. Diagrammatic representation of fuel combustion.

Volumetric analysis. 79 per cent N_2, 21 per cent O_2

Weight analysis. 77 per cent N_2, 23 per cent O_2

Molecular weight. 29 lb per mole

Standard air. 60 per cent relative humidity, 80 F dry bulb, 0.013 lb moisture per lb dry air

Mole relations. 3.76 moles N_2 per mole O_2

For any fuel the dry air theoretically needed is

$$W_a = \frac{W_0}{0.232} = 11.5C + 34.5\left(H_2 - \frac{O}{8}\right) + 4.3S \qquad \text{lb per lb fuel} \qquad (2\text{-}4)$$

The weight of air per unit weight of fuel, the *air-fuel ratio,* plays an important part in combustion calculations.

Supplying only theoretical air amounts in furnaces would not burn the fuel completely. An amount greater than W_a, an additional amount of *excess air,* must be supplied. Practical conditions of fuel type, furnace arrangements, and heat-transfer surface arrangement determine the total amount of air needed for complete combustion.

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* O_2 in fuel.

If W_A is the weight of air *actually* supplied to a furnace per pound of fuel, then

$$\text{Excess-air percentage} = \frac{W_A - W_a}{W_a} \times 100 \qquad (2\text{-}5)$$

$$\text{Dilution coefficient} = \frac{W_A}{W_a} \qquad (2\text{-}6)$$

The volumes of air corresponding to W_A or W_a for any air temperature and pressure are found from ($PV = MRT$)

$$V = \frac{(W_A \text{ or } W_a) \times 53.3 \times (t_a + 460)}{144 \times p_a} \qquad (2\text{-}7)$$

where V = air volume, cu ft
t_a = air temperature, °F
p_a = air pressure, psia

For a gaseous fuel the weight or volume of air can be expressed in terms of 1 cu ft instead of 1 lb of the fuel gas. The previous equations have been developed on the basis of 1 lb of fuel (gas, liquid, or solid), for which the density of the fuel-gas mixture is needed. If not known, it can be obtained from the molal relations by

$$D_M = \frac{MW}{359 \times (14.7/p_M) \times [(t_M + 460)/492]} \qquad (2\text{-}8)$$

where D_M = density of fuel-gas mixture, lb per cu ft
p_M = gas pressure, psia
t_M = gas temperature, °F
MW = molecular weight of gas mixture, lb

To burn a fuel completely, four basic conditions must be fulfilled: we must (1) supply enough *air,* (2) thoroughly *mix* fuel and air, (3) maintain a furnace *temperature* high enough to ignite incoming fuel-air mixture, and (4) provide enough furnace volume to allow *time* for completing the combustion reaction. Gases produced by a burning fuel particle tend to blanket the particle surface and prevent additional oxygen from contacting the unburned portion. Turbulent furnace conditions help to strip away the forming combustion gases continuously and bring up oxygen to the fuel particles. The fuel elements and oxygen molecules react practically instantaneously, but the physical movement of gases and solids requires time. High temperature keeps the reaction going; too much cooling promptly stops the chemical combination of elements.

The following examples illustrate the weight and mole methods of air calculations.

TABLE 2-1. PROPERTIES OF COMMON COMBUSTIBLES AND PRODUCTS

Substance	Formula	Molecular weight	Lb per cu ft, 60 F, 14.7 psia	Cu ft per lb, 60 F, 14.7 psia	Gas, liquid, or solid	Heating value, Btu	
						Per lb	Per cu ft, 60 F, 14.7 psia
Carbon	C	12.00	......	.....	S	14,544	
Hydrogen	H_2	2.02	0.0053	188	G	62,028	325
Sulfur	S	32.07	......	.....	S	4,050	
Carbon monoxide	CO	28.00	0.0739	13.54	G	4,380	323
Methane	CH_4	16.03	0.0423	23.69	G	23,670	1012
Acetylene	C_2H_2	26.02	0.0686	14.58	G	21,500	1483
Ethylene	C_2H_4	28.03	0.0739	13.54	G	21,320	1641
Ethane	C_2H_6	30.05	0.0792	12.63	G	22,070	1762
Benzene	C_6H_6	78.05	0.2061	4.85	G	18,070	3673
Benzene	C_6H_6	78.05	......	.....	L	18,070	
Hexane	C_6H_{14}	86.11	0.2277	4.39	G	20,900	4700
Hexane	C_6H_{14}	86.11	......	.....	L	20,900	
Octane	C_8H_{18}	114.14	......	.....	L	20,580	
Oxygen	O_2	32.00	0.0844	11.84	G		
Nitrogen	N_2	28.02	0.0739	13.52	G		
Air	.....	29	0.0765	13.07	G		
Carbon dioxide	CO_2	44.00	0.1145	8.73	G		
Sulfur dioxide	SO_2	64.07	0.1692	5.91	G		
Water vapor	H_2O	18.02	......	.....	G		

Example 2-3. Find the dry air theoretically needed to burn completely a fuel gas of the following volumetric proportions:

$$CH_4 = 65.0\% \qquad C_2H_6 = 35.0\%$$

Solution. By Eqs. (2-4) and (2-7) and the analysis conversion equations (2-3) we can write

$$MW = 0.65 \times 16 + 0.35 \times 30 = 20.9 \text{ lb per mole}$$

$$C = \frac{12 \times 0.65 + 24 \times 0.35}{20.9} = 77.5\% \text{ by weight}$$

$$H = 100 - 77.5 = 4 \times 0.65 + 6 \times 0.035 = 22.5\% \text{ by weight}$$

$$W_a = 11.5 \times 0.775 + 34.5\left(0.225 - \frac{O}{8}\right) + 4.3 \times 0 = 16.67 \text{ lb per lb}$$

$$V = \frac{16.67 \times 53.3 \times (80 - 460)}{144 \times 14.7} = 226.5 \text{ cu ft of air at 80 F and 14.7 psia per lb gas}$$

By Eqs. (2-1) applied to gas mixtures and writing reaction equations for all combustible components in the gas mixture,

$$CH_4 + 2O_2 = CO_2 + 2H_2O$$
$$2C_2H_6 + 7O_2 = 4CO_2 + 6H_2O$$

Mixture component	Moles per 100 moles gas	Moles O_2 required
CH_4	65.0	$2 \times 65.0 = 130.0$
C_2H_6	35.0	$\frac{7}{2} \times 35.0 = 122.5$
		Total $= 252.5$

$$\text{Moles air} = 252.5 + 3.76N_2 \times 252.5 = 1{,}203 \text{ moles air per 100 moles fuel}$$

$$W_a = \frac{1{,}203 \times 29}{100 \times 20.9} = 16.70 \text{ lb per lb fuel}$$

$$V = \frac{1{,}203}{100} = 12.03 \text{ cu ft air per cu ft gas (both at same temperature and pressure)}$$

Example 2-4. Determine the air-fuel ratio for a liquid fuel whose average composition is represented by the formula $C_{12}H_{26}$ if 30 per cent air is used.

Solution. By Eqs. (2-1) applied to the oil formula

$$2C_{12}H_{26} + 37O_2 = 24CO_2 + 26H_2O$$
$$MW = 12 \times 12 + 26 = 170 \text{ lb per mole of fuel}$$
$$\text{Moles } O_2 \text{ theoretically needed} = {}^{37}\!/\!_2 \text{ moles per mole fuel}$$
$$\text{Moles air} = 1.3({}^{37}\!/\!_2 + 3.76 \times {}^{37}\!/\!_2) = 114.5$$
$$\text{Air-fuel ratio} = \frac{114.5 \times 29}{1 \times 170} = 19.6 \text{ lb per lb fuel}$$

Example 2-5. Given the ultimate analysis of coal,

$$C = 70.5 \quad O = 6.0 \quad S = 3.0 \quad W = 4.0$$
$$H = 4.5 \quad N = 1.0 \quad A = 11.0$$

Determine:

a. Theoretical dry air needed in pounds
b. Volume of actual air for 30 per cent excess air at 60 F and 14.7 psia

Solution. By Eqs. (2-4) and (2-7)

$$a.\ W_a = 11.5 \times 0.705 + 34.5\left(0.045 - \frac{0.068}{8}\right) + 4.3 \times 0.030 = 9.53 \text{ lb per lb}$$

$$b.\ W_A = W_a \times 1.30 = 9.53 \times 1.30 = 12.39 \text{ lb per lb}$$

$$V = \frac{12.39 \times 53.3 \times 520}{144 \times 14.7} = 162.4 \text{ cu ft per lb}$$

The small amount of water vapor in air is frequently neglected in these calculations. When needed, psychrometric data based on wet- and dry-bulb temperatures of the air may be used to calculate the water vapor per pound of dry air.

For many purposes dry air needed for combustion can be calculated from the heating value of the fuel. Coals require about 7.5 lb air for 10,000 Btu in the coal. The constant varies somewhat with the volatile matter and the type of fuel.

2-3. Combustion-gas Products. Complete combustion processes form the gases carbon dioxide (CO_2), sulfur dioxide (SO_2), and water vapor (H_2O). Gases exhausted from power equipment also contain nitrogen

(N_2), oxygen (O_2), and carbon monoxide (CO). The nitrogen comes from the air supply, and the oxygen from the excess air. Carbon monoxide forms with the other gases if all the carbon does not burn completely. Mechanical moisture in the fuel and water vapor in the air reappear in the exhaust gases.

Sometimes combustible elements may pass through a combustion chamber or furnace unburned and pass out with the exhaust gases.

Combustion-gas quantities can be figured on the pound-weight or pound-mole basis. Weights of individual combustion gases can be calculated from Eqs. (2-1), assuming no carbon in the refuse, as in ashes from a boiler furnace. Fuel analysis, excess air needed, and completeness of combustion must be known. Weights of combustion gases based on 1 lb of fuel (solid, liquid, or gaseous) are

$$CO_2 = {}^{44}\!/_{12} \times Y \times C \tag{2-9a}$$
$$CO = {}^{56}\!/_{24} \times (1 - Y) \times C \tag{2-9b}$$
$$O_2 = 0.232 \times X \times W_a + 1.33 \times (1 - Y) \times C \tag{2-9c}$$
$$N_2 = 0.768 \times (1 + X) \times W_a + N \tag{2-9d}$$
$$H_2O = {}^{36}\!/_{4} \times H + W + W_{AV} \tag{2-9e}$$
$$SO_2 = {}^{64}\!/_{32} \times S \tag{2-9f}$$

where $CO_2, CO, O_2, N_2, SO_2, H_2O$ = lb of products per lb of fuel
C, H, S, N = fuel elements, lb per lb of fuel
X = excess air ratio
Y = ratio of C in fuel oxidized to CO_2
W_a = theoretical dry air, lb per lb of fuel [see Eq. (2-4)]
W = mechanical moisture in fuel, lb per lb
W_{AV} = water vapor in air supply = $(1 + X) \times W_a \times G/7{,}000$
G = grains of moisture per lb of dry air from psychrometric chart based on wet- and dry-bulb air temperatures

The total combustion-gas volume and its analysis can be determined. The following equation gives the total moles of gas per pound of fuel:

$$M_t = \frac{CO_2}{44} + \frac{CO}{28} + \frac{O_2}{32} + \frac{N_2}{28} + \frac{SO_2}{64} + \frac{H_2O}{18} \tag{2-10}$$

From this the gas volume is

$$V_G = M_T \times 359 \times \frac{t_g + 460}{492} \times \frac{14.7}{p_g} \tag{2-11}$$

where V_G = total gas volume, cu ft per lb of fuel
M_T = total moles of gas per lb of fuel
t_g = gas temperature considered, °F
p_g = gas pressure considered, psia

The volumetric analysis is given by

$$CO_2,\ \%\ \text{by volume} = \frac{CO_2,\ \text{lb per lb fuel}}{44 \times M_T} \tag{2-12a}$$

and likewise

$$CO = \frac{CO}{28 \times M_T} \tag{2-12b}$$
$$O_2 = \frac{O_2}{32 \times M_T} \tag{2-12c}$$
$$N_2 = \frac{N_2}{28 \times M_T} \tag{2-12d}$$
$$SO_2 = \frac{SO_2}{64 \times M_T} \tag{2-12e}$$
$$H_2O = \frac{H_2O}{18 \times M_T} \tag{2-12f}$$

Combustion gas contains *dry gases* and water vapor. To analyze gases on a dry-gas base, use the same equations, but delete the H_2O item in Eqs. (2-10) and (2-12). The analysis can also be made on a moisture- and sulfur-free basis by eliminating in addition the SO_2 item.

Example 2-6. For a fuel oil with this ultimate analysis, $C = 83.7$, $H = 12.7$, $S = 0.7$, $N = 1.7$, and $O_2 = 1.2$, assume combustion air to have a dry-bulb temperature of 80 F and a wet-bulb temperature of 70 F. With 30 per cent excess air and complete combustion find:

a. Total combustion-gas volume at 400 F and 14.7 psia
b. Dry-gas analysis based on CO_2, O_2, and N_2

Solution. By Eqs. (2-4) and (2-9) to (2-12)

a. $W_a = 11.5 \times 0.837 + 34.5\left(0.127 - \frac{0.012}{8}\right) + 4.3 \times 0.007$
$= 13.98$ lb air per lb oil
$W_{AV} = (1 + X)W_a \times \frac{G}{7{,}000}$
G = 95 grains per lb dry air from psychrometric chart at stated temperatures
$W_{AV} = (1 + 0.30)13.98 \times 95/7{,}000 = 0.3$ lb vapor per lb dry air

Component-gas weights in pounds per pound of fuel are

$$\begin{aligned}
CO_2 &= {}^{44}\!/_{12} \times 0.837 = 3.07\\
CO &= 0\ \text{(complete combustion)}\\
O_2 &= 0.232 \times 0.30 \times 13.98 = 0.97\\
N_2 &= 0.768(1 + 0.30) \times 13.98 + 0.017 = 13.99\\
SO_2 &= {}^{64}\!/_{32} \times 0.007 = 0.02\\
H_2O &= {}^{36}\!/_{4} \times 0.127 + 0.3 = 1.44
\end{aligned}$$

Gas moles per pound of fuel are

$$CO_2 = \frac{3.07}{44} = 0.0698$$

$$O_2 = \frac{0.97}{32} = 0.0303$$

$$N_2 = \frac{13.99}{28} = 0.4992$$

$$SO_2 = \frac{0.02}{64} = 0.0003$$

$$H_2O = \frac{1.44}{18} = \underline{0.0800}$$

$$M_T = \quad 0.6796$$

$$V_G = 0.6796 \times 359 \frac{400 + 460}{492} \frac{14.7}{14.7} = 427 \text{ cu ft per lb oil}$$

b. Gas analysis by volume:

$$CO_2 = \frac{0.0698}{0.0698 + 0.0303 + 0.4992} = \frac{0.0698}{0.5993} = 0.1164 = 11.64\%$$

$$O_2 = \frac{0.0303}{0.5993} = 0.0505 = 5.05\%$$

$$N_2 = \frac{0.4992}{0.5593} = 0.8331 = 83.31\%$$

The mole concept may be used to compute combustion products instead of the weight method. First write a balanced combustion equation; from it find the total gas products and analysis by Eqs. (2-9) and (2-2). Examples 2-3 and 2-4 show how to find air requirements from the reaction equation.

Example 2-7. Write the complete combustion equation for a coal burned in 35 per cent excess air. The ultimate analysis is

$C = 71.7$ $O = 8.3$ $H = 5.2$ $S = 3.4$ $N = 1.3$ $A = 6.6$ $W = 3.5$

Solution. Write the hypothetical mole relations for 100 lb of coal (see Example 2-1).

Element	Weight	Molecular weight	Moles
C	71.7	12	5.98
H	5.2	2	2.6
S	3.4	32	0.11
O	8.3	32	0.26
N	1.3	28	0.05
Total	89.9	. . .	9.00

To burn 100 lb of coal, we must satisfy

$$9.00(C,H,S,O,N) + \left(\frac{3.5}{18}\right)^{*} H_2O + M_0(O_2 + 3.76^{\dagger}N_2) \times 1.35$$
$$= 5.98CO_2 + M_0 \times 0.35O_2 + 0.11SO_2 + 2.79H_2O^{\ddagger}$$
$$+ [(M_0 \times 3.76 \times 1.35) + 0.05]N_2$$

Oxygen weight in fuel and air = oxygen weight in combustion gases

Oxygen entering process:

$$\begin{aligned} 0.26O_2 &= 0.26 \times 32 = 8.3 \\ \frac{3.5}{18} \times H_2O &= 0.19 \times 16 = 3.1 \\ 1.35M_0O_2 &= 1.35M_0 \times 32 = \underline{43.2M_0} \\ \text{Total} &= 11.4 + 43.2M_0 \end{aligned}$$

Oxygen leaving process:

$$\begin{aligned} 0.35M_0O_2 &= 0.35 \times M_0 \times 32 = 11.2M_0 \\ 5.98CO_2 &= 5.98 \times 32 = 191.5 \\ 0.11SO_2 &= 0.11 \times 32 = 3.5 \\ 2.79H_2O &= 2.79 \times 16 = \underline{44.7} \\ \text{Total} &= 239.7 + 11.2M_0 \end{aligned}$$

Equating,

$$11.4 + 43.2M_0 = 239.7 + 11.2M_0$$
$$M_0 = 7.14$$

The complete combustion equation for 100 lb of coal is then

$$9.00(C,H,S,O,N) + 0.20H_2O + 6.6A + 9.64(O_2 + 3.76N_2)$$
$$= 5.98CO_2 + 2.5O_2 + 0.11SO_2 + 2.79H_2O + 36.3N_2 + 6.6A$$

Example 2-8. For the gas with a volumetric analysis of

$CO_2 = 2.4$ $N_2 = 1.4$ $CH_4 = 73.6$ $C_2H_6 = 22.6$

and with dry air in 25 per cent excess find:

a. Total gas volume, for complete combustion, at 500 F and 14.7 psia
b. Dry-gas analysis based on CO_2, O_2, and N_2

Solution. Write the balanced reaction equation for 100 moles of fuel gas by equating masses of elements on both sides.

$$2.4CO_2 + 1.4N_2 + 73.6CH_4 + 22.6C_2H_6 + 226.3(O_2 + 3.76N_2) \times 1.25$$
$$= 121.3CO_2 + 215.0H_2O + 56.6O_2 + 1065.4N_2$$

$$MW \text{ (fuel gas)} = 0.024 \times 44 + 0.014 \times 28 + 0.736 \times 16 + 0.226 \times 30$$
$$= 20.0 \text{ lb per mole}$$

$$\text{Moles air} = \frac{226.3(32 + 3.76 \times 28)1.25}{29} = 1{,}340$$

* From $W = 3.5$.
† From $N_2/O_2 = 0.79/0.21 = 3.76$.
‡ From $3.5/18 + 5.2/2 = 2.79$.

Reaction mole ratios = 100 moles fuel gas + 1,340 moles air $\Rightarrow$ 1,458 moles combustion gas

$$MW \text{ (combustion gas)} = \frac{121.3 \times 44 + 215.0 \times 18 + 56.6 \times 32 + 1{,}065.4 \times 28}{1{,}458}$$

$$= 27.8 \text{ lb per mole}$$

$$M_T = \frac{1{,}458}{100 \times 20.0} = 0.729 \text{ moles of combustion gas per lb fuel}$$

$$V_G = 0.729 \times 359 \frac{960}{492} \frac{14.7}{14.7} = 510 \text{ cu ft per lb fuel}$$

b. Gas analysis by volume:

$$CO_2 = \frac{121.3}{121.3 + 56.6 + 1{,}065.4} = \frac{121.3}{1{,}243.3} = 0.098 = 9.8\%$$

$$O_2 = \frac{56.5}{1{,}243.3} = 4.6\%$$

$$N_2 = \frac{1{,}065.4}{1{,}243.3} = 85.6\%$$

By writing equations as in Examples 2-7 and 2-8 the mole method may be used in combustion calculations for all fuel types.

2-4. Water Vapor. Whenever water vapor is accounted for in combustion calculations, it is added to the left-hand side of the reaction equation and its effect carried over to the right-hand side.

Water vapor in combustion products (usually at a low partial pressure) is treated as a gas. The partial pressure of the vapor, based on Dalton's law, is proportional to the molal percentage or volumetric percentage of the vapor in the total gas products. The *saturation temperature* of steam at this *partial pressure* is the water dew point or temperature at which the water vapor condenses. Sulfur dioxide and oxygen (O_2) in combustion gases react with water vapor to form acids. When gas temperature drops below the dew point, these gases corrode metal surfaces on which they precipitate. Sulfur content, excess air, burner design, furnace temperature, and homogeneity of gases affect sulfur oxide formations. Hence the water dew point must be taken as an index and not as the exact temperature for condensing corrosive acids.

2-5. Combustion Processes. Combustion products are not formed in one simple chemical reaction. Equations like Eq. (2-1) give only the final results of a series of reactions called the "mechanism" of combustion from fuel to final products. Hydrogen and carbon take the form of gaseous hydrocarbons, solid carbon, or mixtures of hydrogen and carbon monoxide in most fuels. Some fuels such as coal gas are simple mixtures of these elementary fuels. In other fuels they may be formed during burning. In oils volatilization produces hydrocarbons; cracking these may yield solid carbon and hydrogen.

Present theories regarding the combustion of solid carbon assume that oxygen penetrates the carbon surface to react and form an initial compound. This eventually becomes carbon monoxide or dioxide, which diffuses from the surface to allow successive penetrations of more oxygen. Hydrogen and carbon monoxide react with oxygen to form intermediate products that are destroyed in later steps. In this chain reaction the intermediate products are called chain carriers.

Chain carriers may exist for short instants, but the simple reaction equations give the end effect. Heating and oxygen affect the burning of hydrocarbons. For rapid heating but retarded mixing with oxygen the hydrocarbons usually crack—they break into carbon and hydrogen, which burn as described above.

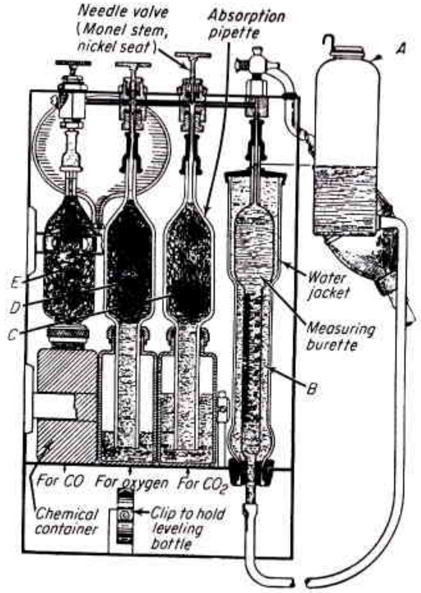

Fig. 2-2. Manually operated Orsat gas analyzer. *A*, leveling bottle; *B*, measuring burette; *C*, *D*, *E*, selective gas-absorbing media. (*Courtesy of The Hays Corporation.*)

Heating and good mixing with oxygen of basic fuels form hydroxylated compounds which break down into a series of compounds called "aldehydes." Under the influence of heat and oxygen the aldehydes break down to form carbon monoxide and hydrogen or water and carbon dioxide. Any carbon monoxide and hydrogen formed burn in a chain-reaction effect. Simple equations like Eq. (2-1) give the integrated effect but no information on the complex intermediate events.

2-6. Flue-gas Analysis. An Orsat analyzer determines the volumetric analyses of gases formed by combustion in a boiler furnace (Fig. 2-2). Leveling bottle *A* induces a flue-gas sample to flow into the measuring burette. Manipulating the bottle enables transferring the sample to contact an absorbing medium *C*, which removes the carbon dioxide from the gas mixture. By transferring the mixture back to the burette the decrease in volume measures the carbon dioxide removed. Successively passing the gas to contact other absorbents allows measuring the oxygen and carbon monoxide removed.

During a test of boiler performance gas samples are analyzed periodically. Gases must be withdrawn from several points simultaneously to get a representative sample. The simple analyzer measures these gases on a volume-fraction basis: CO_2, CO, O_2, and N_2.

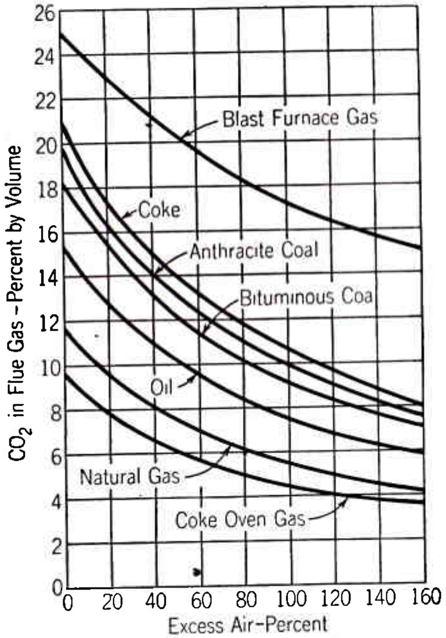

FIG. 2-3. Carbon dioxide (CO_2) variation with fuels and excess air.

Many boiler plants use continuously recording or indicating automatic gas analyzers. Many of these instruments record only the CO_2 or O_2 or unburned fuel-gas percentages. Analyzers cannot determine water-vapor percentages; so their data are on a *dry-gas basis*. When the SO_2 is not removed separately, it becomes absorbed with the CO_2. Air flow and gas flow may be calculated from flue-gas analyses. Depending on the problem, gas samples may be taken from the furnace, furnace exit, boiler passes, or breeching.

2-7. Controlling Excess Air. Proper control of the right amount of excess air maintains optimum combustion efficiency. CO_2 and O_2 in combustion gases are indexes of excess air. The desirable CO_2 level depends on the fuel and optimum excess air for the furnace.

Figure 2-3 shows the variations for complete combustion of a series of fuels. Desirable O_2 values depend much less on the type of fuel, as shown in Fig. 2-4. This makes O_2 measurement the preferred method for furnace control. A series of tests at different steam outputs for a range of excess-air feeds indicates the optimum value of excess air for best efficiency (Fig. 2-5).

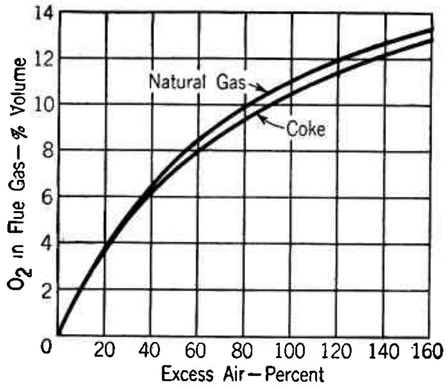

FIG. 2-4. Oxygen (O_2) variation with fuels and excess air.

Boiler losses are computed by methods outlined in Secs. 2-8 and 2-9 for different outputs and CO_2 and O_2 proportions in flue gases. The excess air is then adjusted during normal operations by controlling air feed to show the optimum value of CO_2 or O_2. Smoke formation and slagging of boiler surfaces also play a part in determining the optimum excess air; these are often more important than minimum losses. To

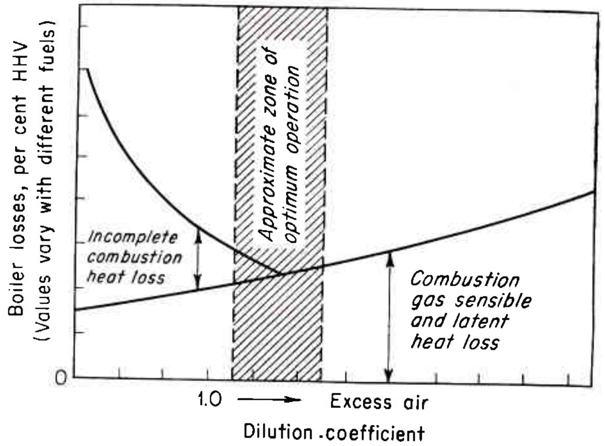

FIG. 2-5. Air flow related to optimum combustion efficiency.

calculate excess air, use

$$\text{Per cent excess air} = \frac{O_2 - 0.5CO}{0.264N_2 - (O_2 - 0.5CO)} \times 100 \qquad (2\text{-}13)$$

where O_2, CO, and N_2 are volumetric percentages in the dry flue gas.

Some instruments meter air and gas flow directly to aid control of air supply. These also indicate steam flow and aid combustion control. Table 2-2 gives ranges of excess air used in power plants.

TABLE 2-2. EXCESS AIR USED IN FURNACE COMBUSTION

Fuel	Equipment	Excess air, per cent
Coal	Stoker-fired	15–65
	Water-cooled furnace, (pulverized coal)	5–40
	Cyclone furnace (crushed coal)	10–15
Fuel oil	Oil burners	5–20
Natural gas	Gas burners	5–12

2-8. Actual Combustion Products and Air Supplied. Burning a fuel produces gases in the furnace. In addition, a solid fuel leaves a solid refuse of ash and some unburned combustible. The refuse may collect in an ashpit, cinder trap, or precipitator.

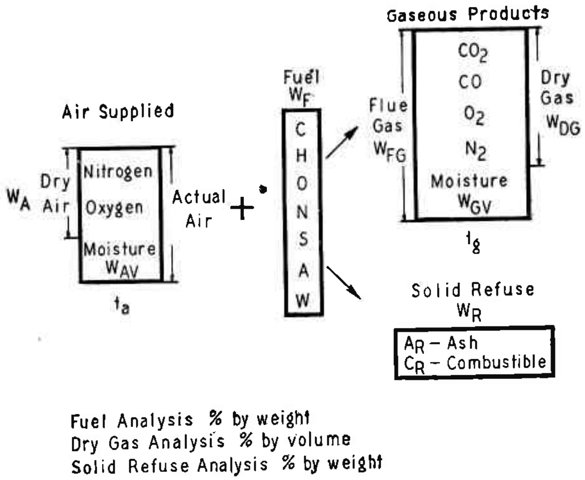

FIG. 2-6. Graphic representation of the combustion of power-plant fuels.

Figure 2-6 diagrams a power-plant combustion process for coal. Liquid and gaseous fuels have no ash component, and their analyses would be different. Refuse, air, and combustion gases are calculated as follows:

Referring to Fig. 2-6, let

W_F = fuel weight, lb (common basis is 1 lb)
W_A = lb of dry air actually supplied to unit per lb of fuel fired, lb
W_{FG} = lb of flue gas per lb of fuel
V_{FG} = cu ft of flue gas per lb of fuel
p_{FG} = flue-gas pressure, psia
t_g = flue-gas temperature, °F
W_{DG} = lb of dry gas per lb of fuel
MW_{DG} = molecular weight of dry gas
W_R = lb of dry solid refuse from W_F lb of fuel
W_{AV} = lb of moisture in W_A lb of dry air
W_{GV} = lb of moisture in W_{FG} lb of flue gas
$C + H + O + N + S + A + W = 1.00$ = fuel analysis by weight fraction
$CO_2 + CO + O_2 + N_2 = 100$ = dry-gas percentage analysis by volume
$A_R + C_R = 1.00$ = dry-refuse analysis by weight fraction
C_F = lb of carbon in dry gas per lb of fuel fired (present in CO_2 and CO)

Refuse calculation:

$$W_R \times A_R = W_F \times A$$

$$\frac{A}{A_R} = \frac{W_R}{W_F} \qquad \text{lb refuse per lb fuel}$$

$$\frac{A}{A_R} C_R = \text{lb of carbon in refuse per lb fuel burned} \qquad (2\text{-}14)$$

Dry-gas calculation:

$$C_F = C - \frac{A}{A_R} C_R \qquad (2\text{-}14a)$$

$$W_{DG} \frac{12(CO_2 + CO)}{44CO_2 + 28CO + 28N_2 + 32O_2} = C_F$$

$$W_{DG} = \frac{11CO_2 + 7CO + 7N_2 + 8O_2}{3(CO_2 + CO)} C_F \qquad (2\text{-}15)$$

But $CO + N_2 = 100 - (CO_2 + O_2)$ from the gas analyzer, and

$$W_{DG} = \frac{4CO_2 + O_2 + 700}{3(CO_2 + CO)} C_F \qquad (2\text{-}16)$$

This is an approximate value, as no allowance exists for the sulfur in the fuel. The precise history of the oxidation of the sulfur is not known; some may not be oxidized and may form part of the refuse. Sulfur dioxide (SO_2) and sulfur trioxide (SO_3) may be formed, reaction with water vapor may occur, or they may eventually discharge with the gases or the solid refuse. Based on simple ratios from Eqs. (2-1) and air data, a given weight of sulfur makes only 0.425 as much dry gas as the same weight of carbon if oxidized to sulfur dioxide. On this basis Eqs. (2-15) can be modified,

$$W_{DG} = \frac{4CO_2 + O_2 + 700}{3(CO_2 + CO)}(C_F + 0.425S) \tag{2-17}$$

Air-supplied calculation:

$$0.77W_A = W_{DG}\frac{28N_2}{44CO_2 + 28CO + 28N_2 + 32O_2} - N \tag{2-18}$$

Assuming N negligible and substituting for W_{DG} from Eqs. 2-15,

$$W_A = \frac{3.04N_2}{CO_2 + CO}C_F \tag{2-19}$$

Based on a mass balance between air, fuel, gas, and refuse, the equation can be written

$$W_A = W_{DG} + 8\left(H - \frac{O}{8}\right) - C_F - S - N \tag{2-20}$$

Flue-gas calculations:

$$W_{GV} = W + 9H + W_{AV} \tag{2-21}$$

$$W_{FG} = W_{DG} + W_{GV} \tag{2-22}$$

$$V_{FG} = \left(\frac{W_{GV}}{18} + \frac{W_{DG}}{MW_{DG}}\right) \times 359 \times \frac{14.7}{p_{FG}}\frac{t_g + 460}{492} \tag{2-23}$$

$$W_A + W_{AV} + 1 \text{ (lb fuel)} = W_R + W_{FG} \tag{2-24}$$

ENERGY BALANCE

2-9. Energy Balance of a Steam Generator. The fuel supplied to a furnace when completely burned releases its heating value. The energy primarily changes the feedwater pumped to the boiler into steam. All the heating value does not go to this useful purpose. There are some losses in the form of incomplete combustion, of flue gases leaving at high temperature, and of radiated and convected heat from the outside of the steam generator. An energy balance shows the distribution of the heating value of the fuel to the formation of steam and to the various losses. In conjunction with the previous symbols let

t_a = entering-air temperature for combustion to the furnace or air heater, if latter is used, °F
t_g = temperature of flue gas after passing all heat-transfer surfaces, °F
h_{g2} = enthalpy of leaving steam, Btu per lb
h_{f2} = enthalpy of water at boiler pressure, Btu per lb
h_{f1} = enthalpy of feedwater, Btu per lb

h_{g4} = enthalpy of reheat steam leaving steam generator, Btu per lb
h_{g3} = enthalpy of reheat steam entering steam generator, Btu per lb
W_S = lb of steam from feedwater per lb of fuel
W_r = lb of reheated steam per lb of fuel
W_B = lb of blowdown per lb of fuel (often negligible)
c_p = specific heat of dry gas, Btu per lb-°F
Δh = enthalpy change of moisture from inlet to flue-gas conditions
HHV = higher heating value of fuel, Btu per lb
Q = various energy values, Btu per lb of fuel as indicated

The individual values usually shown for the heat balance are as follows:

Useful energy:

$$Q_1 = W_S(h_{g2} - h_{f1}) + W_r(h_{g4} - h_{g3}) + W_B(h_{f2} - h_{f1}) \tag{2-25a}$$

Energy loss due to mechanical moisture in fuel:

$$\begin{aligned} Q_2 &= W\,\Delta h \\ \Delta h^* &= (1{,}066 + 0.5t_g - t_a) \qquad \text{when } t_g > 575 \text{ F} \\ &= (1{,}089 + 0.46t_g - t_a) \qquad \text{when } t_g < 575 \text{ F} \end{aligned} \tag{2-25b}$$

Energy loss due to moisture from H in fuel:

$$Q_3 = 9 \times H\,\Delta h \qquad \Delta h \text{ from Eq. (2-25}b\text{)} \tag{2-25c}$$

Energy loss due to moisture from air:

$$Q_4 = W_{AV} \times 0.47 \times (t_g - t_a) \tag{2-25d}$$

Energy loss due to dry-gas sensible heat:

$$Q_5 = W_{DG} \times c_p \times (t_g - t_a) \tag{2-25e}$$

The value of c_p is assumed to be 0.24, the same as for air. Calculations, if warranted, can be made for the mean c_p for the given mixture of dry gases. Heat-loss tables or tables for enthalpies of gases can also be used for this and similar calculations.

Energy loss due to incomplete combustion:

$$Q_6 = W_{DG}\frac{28CO}{44CO_2 + 28CO + 28N_2 + 32O_2} \times 4{,}380 \tag{2-25f}$$

Energy loss due to carbon in refuse:

$$Q_7 = W_R \times C_R \times 14{,}544 \tag{2-25g}$$

Energy loss due to surface heat transfer and unaccounted-for losses:

$$Q_8 = HHV - \Sigma Q_{1\text{ to }7} \tag{2-25h}$$

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* From ASME Power Test Code for Stationary Steam Generating Units, 1946.

The above items for the heat balance are reported in Btu per pound of fuel for each item or as a percentage of the higher heating value. Each heat-balance calculation is for one given load or steam output. The values vary with the rate of steam output, in which case similar items for different outputs are plotted against the outputs, giving the performance curves. These plots generally also include the variation of pressure and temperature of steam, flue gases, and feedwater for the steam generator which come from direct test measurements. An average range of heat-balance values expressed in per cent is

$$Q_1 = 50\text{–}90 \qquad Q_5 = 6\text{–}30 \qquad Q_7 = 0\text{–}2$$
$$Q_{2+3+4} = 3\text{–}\ 5 \qquad Q_6 = 0\text{–}\ 5 \qquad Q_8 = 2\text{–}9$$

The form of calculation outlined above requires weights of fuel and water to be determined. The items can also be calculated by a "heat-balance method." That is, if a value for Q_8, the radiation loss, is assumed, the other items can be evaluated without weighing fuel and water. The accuracy of course depends on how well this loss is known.

The performance of a steam generator is indicated by its efficiency:

$$\eta_b = \frac{Q_1}{HHV} \tag{2-26}$$

This is the *over-all gross efficiency*. If the auxiliary power needed for operating the unit is appreciable, the *net efficiency* may be calculated. The auxiliaries include draft fans, pulverizers, and other energy-consuming equipment. These may be either steam- or motor-driven. In computing net efficiency the ASME Power Test Code indicates arbitrary methods of debiting the Q_1 item to account for energy used by the auxiliaries. Net efficiency may also basically be computed by replacing the W terms in Eq. (2-25*a*) by $W - W_{aux}$ to compute Q_1 and using Eq. (2-26). The term W_{aux} is the weight of steam directly used for steam-driven auxiliaries plus the steam equivalent of the power used by the motor-driven auxiliaries. Net efficiency is more indicative of boiler performance and differs by some 2 to 3 per cent from gross. The gross efficiency is mostly used.

The efficiency of steam generators is first a function of design. Beyond that, efficiency depends upon the loading and manner of operation. Good operation consists in: (1) control and limiting of excess air; (2) minimizing combustibles in refuse; (3) minimizing combustibles in flue gas; and (4) maintaining clean heat-transfer surfaces.

PROBLEMS

2-1. A combustion process occurs between 50 lb of carbon and two-thirds of the air needed for complete combustion. Calculate:

a. Relative weights of carbon dioxide and carbon monoxide produced
b. Percentage of complete-combustion heat developed

2-2. Calculate the relative weights of air needed for the combustion of equal weights of carbon, hydrogen, and sulfur.

2-3. Derive or demonstrate the following equations: (2-2); (2-4); (2-9*a*) to (2-9*f*); (2-13); (2-20); (2-24).

2-4. A producer gas has the following percentage volumetric composition:

$$CO_2 = 10.8 \qquad CO = 18.3 \qquad CH_4 = 3.1$$
$$O_2 = 0.1 \qquad H_2 = 12.9 \qquad N_2 = 54.6$$
$$C_2H_4 = 0.2$$

Calculate:
a. Weight analysis by elements
b. Weight analysis by compounds

2-5. An anthracite coal has the following composition:

$$C = 84.7 \qquad N = 1.5 \qquad A = 5.8$$
$$H = 2.9 \qquad S = 0.8 \qquad W = 2.7$$
$$O = 1.6$$

Calculate for complete combustion per pound of coal:
a. Theoretical weight of oxygen required
b. Actual air needed if 35 per cent in excess

2-6. A bituminous coal has the following composition:

$$C = 71.6 \qquad N = 1.3 \qquad A = 9.1$$
$$H = 4.8 \qquad S = 3.4 \qquad W = 3.5$$
$$O = 6.3$$

Calculate for complete combustion per pound of coal:
a. Theoretical weight of air required
b. Actual weight of air required for a dilution coefficient of 1.6
c. Actual volume of air at 14.7 psia and 80 F

2-7. Calculate the volume of air at 14.6 psia and 70 F required for the complete combustion of the following fuel oil if 20 per cent in excess is used:

$$C = 83.65 \qquad S = 0.75 \qquad O = 1.20$$
$$H = 12.70 \qquad N = 1.70$$

2-8. A natural gas has the following percentage volumetric composition:

$$CH_4 = 59.8 \qquad N_2 = 2.2$$
$$C_2H_6 = 37.6 \qquad CO_2 = 0.4$$

Calculate for complete combustion:
a. Pounds of air theoretically required per pound of gas and per cubic foot of gas at 16.8 psia and 80 F
b. Ratio of air to gas volume at 14.7 psia and 60 F if air is 25 per cent in excess

2-9. A subbituminous coal has the following analysis:

$$C = 60.1 \qquad N = 1.1 \qquad A = 5.0$$
$$H = 4.2 \qquad S = 1.3 \qquad W = 12.2$$
$$O = 16.1$$

Calculate for complete combustion and dry air 40 per cent in excess:

a. Weight of air required per pound of coal

b. Volumetric analysis of the flue gas and dry gas

2-10. A fuel oil has the following weight per cent analysis

$$C = 85.4 \qquad O = 2.8 \qquad S = 0.3$$
$$H = 11.3 \qquad N = 0.2$$

Determine:

a. Hypothetical mole formula for 100 lb of fuel

b. Complete reaction equation per 100 lb of fuel for dry air 40 per cent in excess

c. Dry-gas analysis based on CO_2, O_2, N_2

d. Water dew-point temperature of the flue gas at atmospheric pressure

2-11. A fuel gas has the following volumetric analysis:

$$CH_4 = 68.0 \qquad C_2H_6 = 32.0$$

Assume complete combustion with 15 per cent excess air at 14.7 psia, wet-bulb 70 F and dry-bulb 80 F.

Determine:

a. Volume of actual (dry + water vapor) air required per 1,000 cu ft of gas at same pressure and temperature

b. Water dew-point temperature of the flue gas

2-12. A bituminous coal from Pike Co., Ind., as in Table 1-2 is burned with 35 per cent excess air at 90 F dry-bulb and 80 F wet-bulb.

Determine:

a. Air (dry + water vapor)-fuel ratio

b. Weight of flue gas per pound of coal if dry refuse shows 80 per cent ash

2-13. A bituminous coal has the following analysis:

$$C = 79.6 \qquad N = 1.8 \qquad A = 8.6$$
$$H = 4.5 \qquad S = 0.5 \qquad W = 1.2$$
$$O = 3.8$$

Calculate the dry-gas analysis (CO_2, CO, O_2, N_2) for a dilution coefficient of 1.35, assuming that 5 per cent of the carbon burns to carbon monoxide.

2-14. Air having a dry-bulb temperature of 78 F and a wet-bulb of 68 F is used in burning the following coal:

$$C = 58.0 \qquad N = 1.0 \qquad A = 8.0$$
$$H = 4.8 \qquad S = 2.0 \qquad W = 10.0$$
$$O = 16.2$$

Calculate for complete combustion:

a. Percentage of moisture in the air

b. Pounds of actual air (dry + water vapor) theoretically required

2-15. The analysis of a fuel oil is as follows:

$$C = 83.9 \qquad S = 2.2 \qquad N = 0.0$$
$$H = 13.9 \qquad O = 0.0$$

Calculate for complete combustion with 20 per cent excess of dry air:

a. Flue-gas volume in cubic feet per pound of oil at 400 F and 14.5 psia

b. Dry-gas volumetric analysis (CO_2, O_2, N_2)

2-16. A fuel-gas mixture has the following volumetric analysis:

$$CH_4 = 75.5 \qquad C_2H_6 = 24.5$$

Calculate the dry-gas analysis for complete combustion with dry air if 20 per cent in excess.

2-17. An Orsat apparatus showed the readings given in Table P-1 for each of three similar samples:

TABLE P-1

CO_2	85.8	86.8	86.1
O_2	82.2	82.7	82.4
CO	80.8	81.6	81.1

Calculate the average dry-gas analysis.

2-18. Convert the analysis to the weight basis for both elements and compounds in Prob. 2-17.

2-19. The following gas analysis was obtained for a flue-gas sample:

$$CO_2 = 11.1 \qquad CO = 0.6 \qquad N_2 = 80.5 \qquad O_2 = 7.8$$

Estimate the excess air being used.

2-20. A test at a power station showed 8,133 lb of coal fired per hour. Coal analysis shows an ash percentage of 7.80, and the refuse shows 4.3 per cent combustible.

Calculate:

a. Pounds of refuse per hour

b. Carbon in refuse in pounds per hour

2-21. The analysis of the refuse from a coal-fired furnace which has been sprayed with water is by weight

$$\text{Ash} = 75.0 \qquad \text{Combustible} = 15.0 \qquad \text{Moisture} = 10.0$$

Assume the coal a semibituminous, as in Table 1-2, from Pitkin Co., Colo. Determine the carbon in dry gas per pound of coal.

2-22. The following stack gas was obtained from a furnace burning bituminous coal:

$$CO_2 = 12.6 \qquad CO = 0.2 \qquad N_2 = 80.7 \qquad O_2 = 6.5$$

The ashpit refuse contains 8.5 per cent combustible matter, and the coal used has the following analysis:

$$C = 73.4 \qquad N = 1.5 \qquad A = 7.6$$
$$H = 5.0 \qquad S = 3.0 \qquad W = 3.5$$
$$O = 6.0$$

Calculate:

a. Pounds of dry gas per pound of coal

b. Pounds of flue gas per pound of coal, assuming dry combustion air

2-23. Test data from a boiler are as follows:

Coal analysis:

$$C = 75.0 \qquad N = 2.0 \qquad A = 9.0$$
$$H = 5.0 \qquad S = 1.0 \qquad W = 1.0$$
$$O = 7.0$$

Dry-gas analysis:

$$CO_2 = 12.3 \qquad O_2 = 6.6 \qquad N_2 = 80.8 \qquad CO = 0.3$$

Refuse analysis:

Ash = 92.0

Air has 1.7 per cent moisture.

Calculate per pound of coal:

a. Actual air used

b. Air theoretically needed with given moisture

c. Percentage of excess air and dilution coefficient

2-24. A Texas natural gas has the following percentage volumetric analysis:

$CO_2 = 0.2$ $C_2H_6 = 2.9$ $N_2 = 2.3$
$CH_4 = 94.4$ $O_2 = 0.2$

Dry-gas analysis:

$CO_2 = 10.1$ $O_2 = 5.8$ $N_2 = 84.1$

Calculate, per pound of gas:

a. Pounds of dry gas

b. Pounds of dry air stoichiometrically required

c. Pounds of dry air actually used

d. Percentage of excess air

2-25. A bituminous coal from Jefferson Co., Ohio (Table 1-1), is burned in a furnace with the following results:

$CO_2 = 11.0$ $O_2 = 7.1$ $N_2 = 80.6$ $CO = 1.3$

Refuse analysis:

Ash = 95.0%

Assume the combustion air to be dry, and calculate:

a. Pounds of flue gas per pound of coal

b. Cubic feet of flue gas per pound of coal at 500 F and 14.5 psia

2-26. Data from a test on No. 5 boiler at the River Rouge plant of the Ford Motor Company are as follows:

Ultimate analysis of coal as fired:

$C = 78.0$ $H = 4.92$ $O = 6.76$ $W = 1.70$
$N = 1.60$ $S = 0.62$ $A = 6.40$

HHV of coal as fired, 13,713 Btu per lb
Coal rate, 73,530 lb per hr
Refuse rate and analysis:
Ashpit refuse, 941 lb per hr,

$C = 0.3$ $A = 99.7$

Fly ash, 5,136 lb per hr,

$C = 26.7$ $A = 73.3$

Dry flue-gas analysis:

$CO_2 = 13.7$ $O_2 = 4.5$ $CO = 0.00$ $N_2 = 81.8$

Flue-gas temperature, 375 F
Steam data:
Pressure, 1,260 psia
Temperature, 900 F
Steam rate, 820,000 lb per hr

Feedwater data:
Temperature, 393 F
Pressure, 1,500 psia
Air data:
Dry bulb, 99 F
Wet bulb, 66 F

Calculate the complete energy balance on the Btu per pound of fuel and percentage basis.

2-27. Data from a test on the stoker-fired steam generator of the Federal Central Heating Plant in Washington, D.C., are as follows:

Ultimate analysis of coal as fired:

$C = 76.4$ $H = 4.5$ $O = 2.6$ $W = 3.7$
$N = 1.3$ $S = 1.4$ $A = 10.1$

HHV of coal as fired, 13,410 Btu per lb
Refuse analysis:

$C = 12.2$ Ash = 87.8

Coal rate, 19,812 lb per hr
Dry flue-gas analysis:

$CO_2 = 15.73$ $O_2 = 3.13$ $CO = 0.03$ $N_2 = 81.11$

Flue-gas temperature, 603 F
Steam data:
Pressure, 200 psia
Quality, 99.5 per cent
Steam rate, 194,561 lb per hr

Feedwater data:
Temperature, 122 F
Air data:
Temperature, 60 F
Relative humidity, 40 per cent

Calculate the complete energy balance on the Btu per pound of fuel as fired and percentage basis. Illustrate the balance graphically to some convenient scale.

2-28. Data from a boiler test are as follows:

Ultimate analysis of coal as fired:

$C = 57.61$ $H = 3.73$ $O = 5.80$ $W = 12.00$
$N = 1.07$ $S = 3.29$ $A = 16.50$

HHV of coal as fired, 10,420 Btu per lb
Refuse analysis:

$C = 20.0$ $A = 80.0$

Dry flue-gas analysis:

$CO_2 = 13.0$ $O_2 = 7.0$ $CO = 0.0$ $N_2 = 80.0$

Flue-gas temperature, 350 F
Air data:

Dry bulb, 70 F Wet bulb, 60 F

Calculate an energy balance assuming an "unaccounted and radiation loss" of 3.7 per cent.

2-29. Data from a test on a reheat steam-generating unit are as follows:

Ultimate analysis of coal as fired:

$C = 71.9$ $O = 5.3$ $S = 2.3$ $W = 5.0$
$H = 4.8$ $N = 1.3$ $A = 9.4$

Fuel heating value, dry (laboratory basis), 12,940 Btu per lb
Refuse analysis:

$C = 4.0$

Fuel rate, 80,000 lb per hr
Dry flue-gas analysis:

$CO_2 = 15.9 \qquad CO = 0.0$
$O_2 = 2.6 \qquad N_2 = 81.5$

Flue-gas temperature, 285 F
Steam data:
  Superheater outlet = 2,075 psia, 1050 F, 845,391 lb per hr
  Reheater inlet = 362 psia, 653 F, 691,266 lb per hr
  Reheater outlet = 342 psia, 990 F
Feedwater data:
  Feedwater inlet, 537 F, 2,163 psia
Air data:
  Ambient temperature, 82.7 F dry bulb, 58.9 F wet bulb
  Temperature air for combustion to air heater, 121 F

Calculate an energy balance.

2-30. Ten pounds of steam at 450 psia and 500 F is formed from feedwater at 250 F per lb of coal fired in a steam generator. The refuse contains 9.1 per cent combustible, and the coal has 10.8 per cent ash and a high heat value of 14,150 Btu per lb. The auxiliaries of the steam generator require 4 per cent of the steam generated.

Calculate:

a. Gross steam-generator efficiency
b. Net steam-generator efficiency

2-31. A steam turbine requires 12.0 lb of steam for each kilowatthour of output. Steam conditions are 900 psia and 570 F. Feedwater temperature is 300 F. Three per cent of the turbogenerator output is used for steam-generator auxiliaries. Steam is generated at 12.5 lb of coal. The HHV of coal is 13,610 Btu per lb.

Calculate:

a. Gross efficiency of the steam generator
b. Net efficiency of the steam generator

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CHAPTER 3

ENERGY CYCLES

3-1. Cycles to Convert Energy. Energy as a *heat flow* may originate in the burning of fossil fuels, solar radiation, or nuclear reactions. Space heating and industrial process heating use the greatest amount of energy. *Mechanical energy* appearing as a force acting through a distance or a torque acting through an angle to move pistons or turn shafts of pumps, fans, and generators is as important as heating, but usually smaller in quantity in commercial applications.

The terms *heat* and *work* designate the two fundamental forms of energy flow. Thermal power stations convert heat to work and then often to electrical energy through some form of energy-conversion cycle. The field of thermodynamics encompasses the study of energy cycles.

Heat flow involves a *quantity* aspect and a *temperature pattern.* For example, 970.3 Btu vaporizes 1 lb of water at 14.7 psia and 212 F. The 970.3 Btu is the quantity aspect, and the temperature pattern has no change for the water or steam. Temperature patterns may rise or fall with heat flow into and out of a medium.

The maximum work, or *available energy,* developed from a heat flow depends on the heat quantity, its temperature pattern, and the lowest ambient or receiver temperature available. A heat cycle receives a heat flow at a given temperature pattern, to convert part to work and discard the remainder to the receiver.

Ideal heat cycles use heat in ideal, or reversible, processes of working fluids to develop the available energy as work. *Actual* heat cycles use heat in practical and imperfect processes of working fluids that develop less than the ideal available energy as work and discard more energy to the receiver than the ideal cycle.

Figure 3-1 shows a general heat cycle on temperature-entropy coordinates. Area 4-1-*b*-*a* measures the heat flow into the working fluid that receives it at the temperature pattern 4-1. Area 1-2-3-4 measures the available energy referred to the receiver temperature. Actual temperature differences needed for heat transfer and imperfect processes may give an imperfect cycle like 1-2′-3′-4. The greater area under 2′-3′ compared with that beneath 2-3 shows more heat rejected from the cycle.

This means that the same amount of heat develops less work output in an actual cycle than the ideal cycle.

If process 4-1 were at constant temperature, the cycle 1-2-3-4 would be the Carnot cycle with a thermal efficiency

$$\eta_t = \frac{\text{work output}}{\text{heat received}} = \frac{T_1 - T_2}{T_1} \tag{3-1}$$

where T_1 is the constant source temperature and T_2 the constant receiver temperature. Note that this applies *only* to the ideal Carnot cycle. It is important because it shows that efficiency depends on the temperature of the heat flows into and out of the cycle.

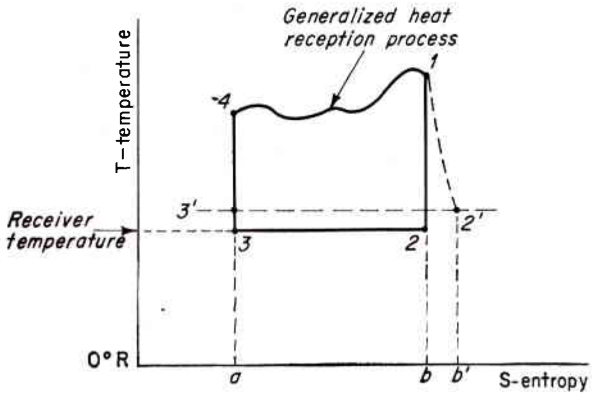


FIG. 3-1. Thermodynamic analysis of heat quantity and temperature pattern of heat flows in heat cycles. Area 4-1-*b*-*a*, heat received; curve 4-1, temperature pattern of heat received; area 2-3-*a*-*b*, unavailable energy (ideal); area 1-2-3-4, available energy (ideal).

Practical heat cycles must consider:

1. *Heat source.* Combustion, fission, fusion, solar radiation, etc.
2. *Cycle fluid.* Water, steam, gases, or any liquid or vapor
3. *Mechanical equipment.* Apparatus in which cycle processes take place
4. *Cycle processes.* Nature and number involved
5. *Heat receiver.* Region or medium with lowest available temperature
6. *Thermal efficiency* = (net work output)/(source heat input)

To measure how perfect actual processes are compared with their ideal counterparts, the following ratios are used:

$$\eta_e = \frac{\text{actual work}}{\text{isentropic work}} \tag{3-2}$$

This is the isentropic engine, or expansion, efficiency, used for work-producing processes.

$$\eta_c = \frac{\text{isentropic work}}{\text{actual work}} \tag{3-3}$$

This is the isentropic compression efficiency, used for work-absorbing processes. Sometimes machine efficiency designates joint compression and engine efficiencies in a cycle.

Most cycles have positive and negative work internal exchanges, the difference usually being the net work output; they are compared by the work ratio

$$r_w = \frac{\text{net cycle work}}{\text{positive cycle work}} \tag{3-4}$$

In a gas-turbine cycle this would be figured as

$$r_w = \frac{\text{turbine output} - \text{compressor input}}{\text{turbine output}}$$

Low work ratios indicate the need for relatively large machines to develop a given net work output. This means a larger investment in equipment.

VAPOR POWER CYCLES

3-2. Rankine Cycle. The Rankine cycle is the simplest theoretical cycle using vapor as the working medium. Figure 3-2*a* and *b* shows the flow diagram and *T-S* diagram of the cycle. Only very small-capacity plants would actually use this cycle. We study it because it gives a simple basis for finding the effects of varying boiler pressure, temperatures, and condenser back pressures or vacuums.

For the ideal *T-S* diagram the area 1-2-3-4 represents the net work output W and the area *a*-3-2-*b* the heat Q_r rejected in the condenser. The area *a*-4-1-*b* represents the total heat added Q_a. The Rankine cycle efficiency is

$$\eta_{rk} = \frac{W}{Q_a} = \frac{W_e - PW}{Q_a} = \frac{Q_a - Q_r}{Q_a} = \frac{(h_1 - h_4) - (h_2 - h_3)}{h_1 - h_4} \tag{3-5}$$

Lowering the condenser pressure will raise the thermal efficiency. The lowest condenser pressure that can be reached ideally will be the saturation pressure corresponding to the lowest available temperature of water or air that acts as the receiver.

Since Q_r can be transferred only through a temperature drop, the actual condenser pressure will always be higher than the theoretical. Figure 3-3 shows the variation of efficiency with condenser back pressure for different initial steam conditions. Actual back pressures depend on cooling-water temperature and flow rate, the amount of heat-transfer surface, and its cleanliness.

Great strides are being made in raising steam-plant thermal efficiency by raising both boiler steam pressures and temperatures. Equation (3-1) shows that cycle efficiency depends only on the temperatures of heat addition and rejection. Specifically, adding the cycle heat at the highest possible average temperature will raise cycle efficiency as much as possible. The average steam temperature in the steam cycle rises by superheating or by raising the boiler pressure or both. Figure 3-4 shows the variation in Rankine-cycle efficiency with steam temperature and pressure for a given value of condenser pressure.

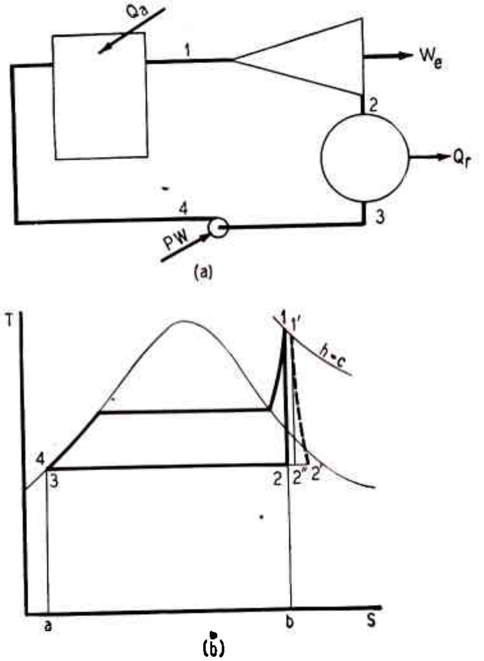


FIG. 3-2. Schematic of simple Rankine cycle. Note throttling $(h = c)$ and actual turbine process 1′-2′. Scale of pump process 3-4 is too small to show on this diagram.

Using high pressures depends upon adequate joints and sealing devices at various points in the cycle. Welding has greatly aided the practicality of high-pressure steam. Steam entering a turbine must have a high enough temperature to avoid having more than 12 per cent moisture in the exhaust. The moisture entrained as a fine mist has great eroding power and rapidly wears away the turbine buckets in the last few stages. For a given initial pressure the higher the initial temperature, the lower the exhaust moisture. Irreversible effects reduce moisture content during expansion.

Using high steam temperatures depends on the availability of steel alloys that can withstand high pressures at these elevated temperatures. Metals *creep* (slowly expand) with a loss of strength when subjected to high temperatures for extended periods. The highest temperature to be applied in steam cycles will be 1200 F, and many commercial cycles use 1050 F.

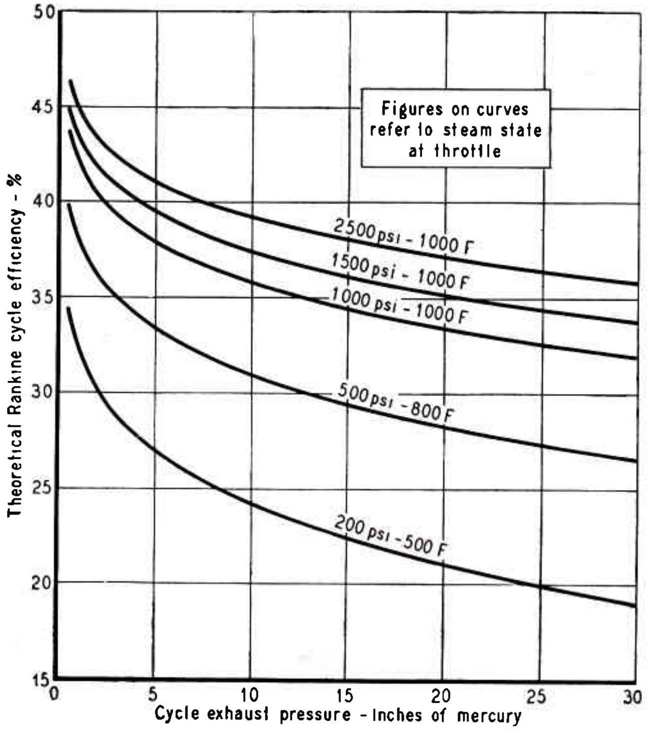


FIG. 3-3. Effect of back pressure on Rankine-cycle efficiency. Curves are for ideal cycle.

The thermal efficiency of an actual plant running on the Rankine cycle is less than the theoretical for given throttle and exhaust conditions. Irreversible effects in the steam main from boiler to prime mover and in the prime mover decrease W_e, correspondingly increasing Q_r. The pump work PW will be greater than ideal because of the pump irreversibility and the losses in the driving engine or motor.

An actual plant has auxiliary devices needing an energy input in addition to the boiler feed pump. This either increases Q_a or decreases W.

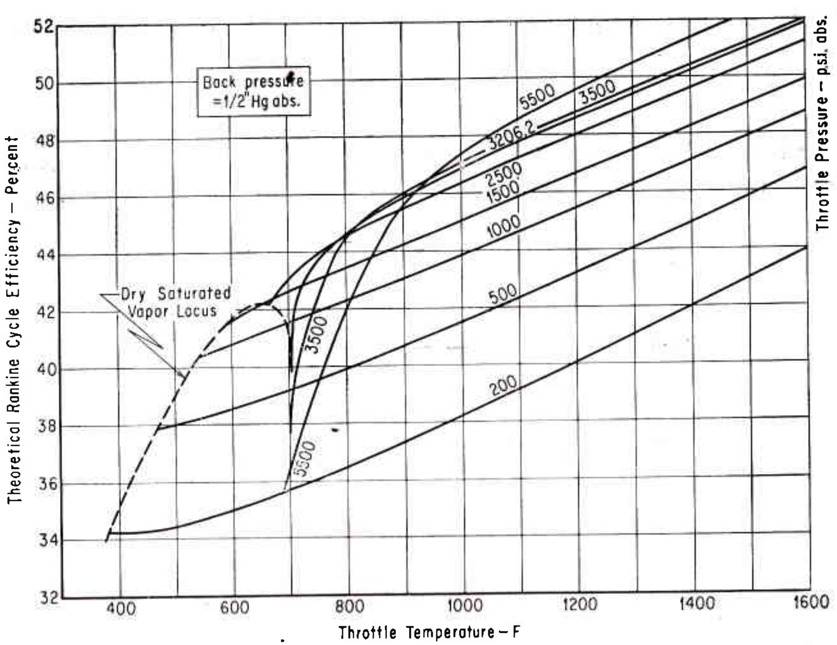


FIG. 3-4. Effect of throttle steam conditions on Rankine-cycle efficiency. Note supercritical curves.

Also, energy input to the actual plant is greater than Q_a by the extent of the boiler losses.

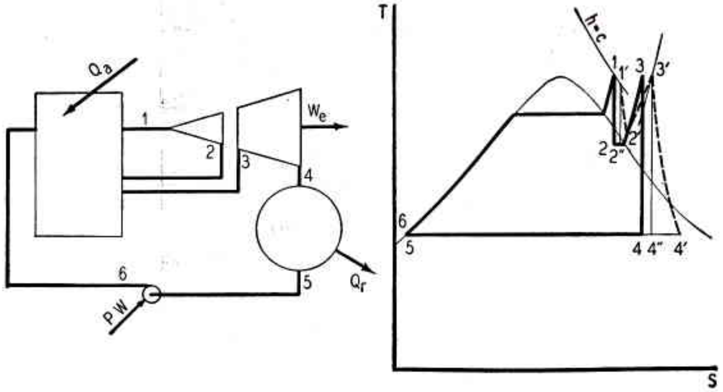
For a nonextraction turbine plant in which the prime mover drives a generator and all the auxiliaries are driven by electric motors supplied from the generator the actual over-all station thermal efficiency will be

$$\eta'_{rk} = \frac{\eta W_e - AW}{Q_a/\eta_b} = \frac{\eta(h_1 - h_{2''}) - AW}{(h_1 - h_4)/\eta_b} \tag{3-6}$$

where η = over-all engine efficiency of prime mover
W_e = theoretical energy = $h_1 - h_{2''}$, Btu per lb
AW = total energy demand of all auxiliary drives, including PW, Btu per lb
η_b = over-all steam-generator efficiency
Q_a = energy added to steam in steam generator = $h_1 - h_4$, Btu per lb
$h_{2''}$ = final enthalpy for isentropic expansion from $h_{1'}$, Btu per lb

3-3. Reheat Cycle. Until about 1920, throttle conditions were normally designed for about 250 psi and 600 F. The limit on the temperature, owing to metallurgical considerations, meant that increasing boiler pressures would materially increase the moisture in exhaust steam above the safe maximum of 12 per cent. To attain the higher thermal efficiencies possible with increased steam pressures, because of higher saturation temperature, the reheat cycle (Fig. 3-5) was created. The simplest form of reheat cycle compared with the Rankine cycle merely means withdrawing the steam from the turbine at some intermediate stage, returning it to the steam generator for resuperheating, and reintroducing it into the turbine at the following stage. There is an optimum reheat pressure for a given initial pressure and temperature (Fig. 3-6).

Though the basic reason for reheating was to make the higher efficiency from increased initial pressure (saturation temperature) attainable, reheating of itself causes a gain in thermal efficiency, as evidenced by the curves at constant initial pressure (Fig. 3-6). This gain is realized because reheating raises the average temperature at which heat is added to the cycle. Today's high-pressure and -temperature plants running at pressures of 1,400 to 2,400 psia and 950 to 1150 F temperatures use reheating as standard practice. The few supercritical plants at 3,500 psia and above and 1050 to 1200 F use *double* reheating for the gain it offers in thermal efficiency.



(a) (b)

FIG. 3-5. Schematic of simple reheat cycle.

In flowing through reheater piping the steam pressure drops (Fig. 3-5), but to ensure proper steam distribution among the many parallel pipes of the reheater the steam pressure may be dropped as much as 75 psi. A reheater is a separate superheater placed in the gas passes of a boiler or steam generator.

The ideal thermal efficiency of the reheat cycle is

$$\eta_{rh} = \frac{W}{Q_a} = \frac{W_e - PW}{Q_a} = \frac{(h_1 - h_2) + (h_3 - h_4) - PW}{(h_1 - h_6) + (h_3 - h_2)} \tag{3-7}$$

The actual over-all station thermal efficiency of a plant operating on a simple reheat cycle is

$$\eta'_{rh} = \frac{\eta_e W_e - AW}{\dfrac{Q_a}{\eta_b}} = \frac{\eta'_e(h_{1'} - h_{2''}) + \eta''_e(h_{3'} - h_{4''}) - AW}{[(h_1 - h_6) + (h_{3'} - h_{2'})]/\eta_b} \qquad (3\text{-}8)$$

where η_e = over-all engine efficiency of first section of turbine from 1 to 2

η''_e = over-all engine efficiency of second section of turbine from 3 to 4

$h_{4''}$ = final enthalpy for isentropic expansion from $h_{3'}$, Btu per lb

$h_{2''}$ = final enthalpy for isentropic expansion from $h_{1'}$, Btu per lb

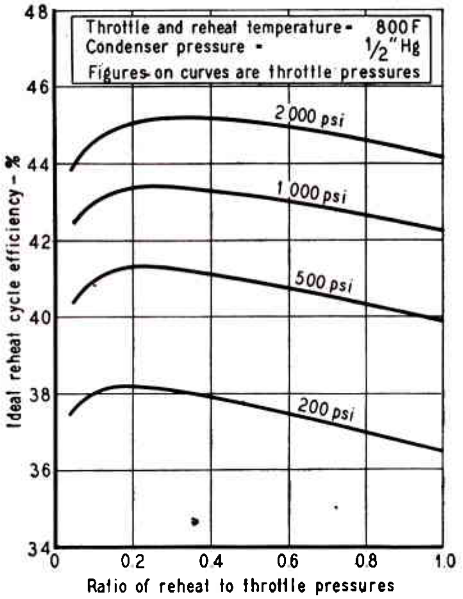


FIG. 3-6. Reheat-cycle efficiency variation with reheat pressure. Curves are for ideal cycle.

3-4. Regenerative Cycle.

A regenerative cycle keeps a given quantity of energy circulating within the cycle; this contributes to decreasing the irreversibility of some processes. The major irreversible process of the Rankine cycle is the mixing of the cold condensate with the saturated steam and water mixture in the boiler. If the condensate could be heated by absorbing heat reversibly from the steam as it expands in the prime mover, the irreversible mixing would be avoided.

Bleeding steam from one or more stages of the turbine to heat the condensate on its way back to the boiler approximates the reversible heating. Figure 3-7*a* shows such an arrangement for a two-heater regenerative cycle. Figure 3-7*b* shows the *T-S* diagram.

The *T-S* diagram is so drawn that the net area 1-4-*a*-*b*-*c*-*d*-10 is equivalent to the ideal W_e of the turbine. The actual-state points of the liquid are from 5 to 10 as indicated, when *S* is the entropy of the steam or water. If m_1 and m_2 are the rates of steam extraction for 1 lb entering the throttle, then there is 1 lb flowing from 1 to 2, $1 - m_1$ lb from 2 to 3, and $1 - m_1 - m_2$ lb from 3 to 4 and into the condenser. The ideal thermal efficiency of a two-heater regenerative cycle is

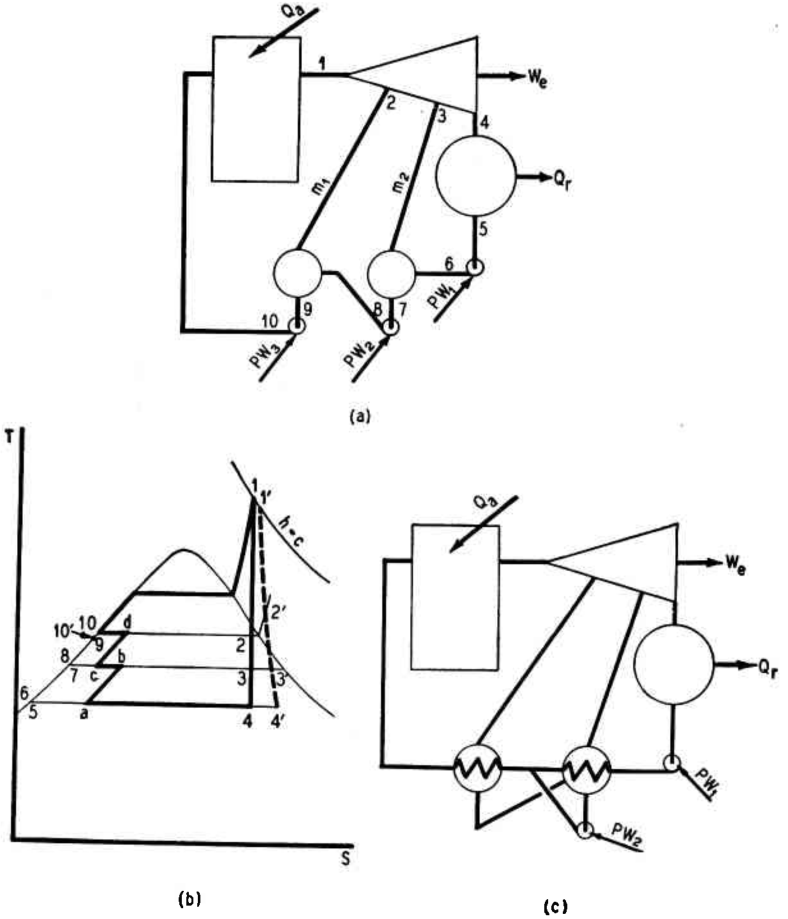


FIG. 3-7. Schematic of regenerative cycle. *T-S* diagram plotted so that enclosed cycle area represents net work.

$$\eta_{rg} = \frac{W}{Q_a} = \frac{W_e - PW}{Q_a}$$

$$= \frac{(h_1 - h_2) + (1 - m_1)(h_2 - h_3) + (1 - m_1 - m_2)(h_3 - h_4) - PW}{h_1 - h_{10}} \qquad (3\text{-}9)$$

where m_1 and m_2 are determined from heater energy balances. Compared with the simple Rankine cycle 1 lb of steam will not produce as great an amount of W_e for the same steam conditions, because of the extractions m_1 and m_2. The marked reduction in Q_a, however, greatly increases the cycle efficiency. For given initial steam conditions the efficiency of the cycle increases with each additional heater. The bleed points are efficiently placed when the enthalpy rise of the condensate in each heater equals the enthalpy difference of saturated water in the condenser and the boiler divided by the number of heaters plus 1. Figure 3-8 shows the increase in theoretical cycle efficiency for given throttle conditions as additional heaters are provided in an ideal regenerative cycle. The zero heater condition is the Rankine cycle; the greatest increment in efficiency occurs by the addition of the first heater; the increments for each additional heater thereafter successively diminish until it is evident that the cost of adding another heater and pump will not in all likelihood be justified by the saving in Q_a. Five points of extraction are mostly used in practice; some cycles use as many as nine.

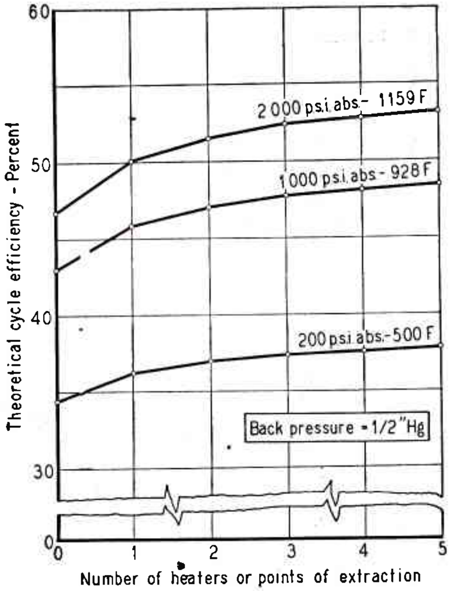
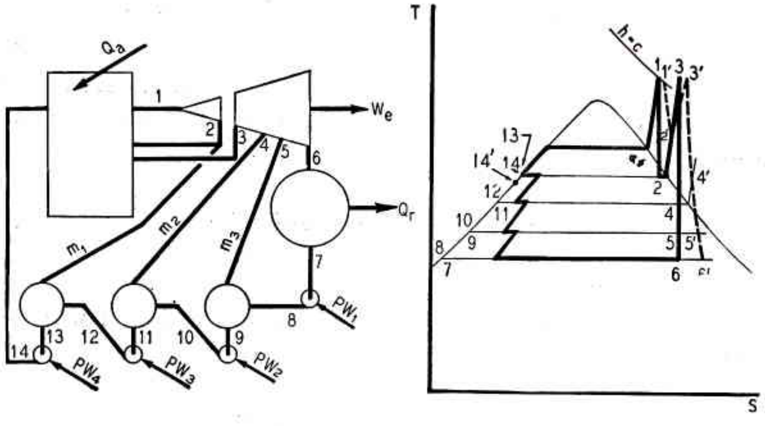


FIG. 3-8. Regenerative-cycle efficiency variation with number of heaters. Curves are for ideal cycle.

To save the cost of pumps, the cascading type of heater installation has often been used (Fig. 3-7*c*). The heaters are all of the closed type instead of the open, or contact, type as in Fig. 3-7*a*. The bleed steam condensate from the high-pressure heater collects in the heater hot well and flows to the lower-pressure heater, where part of this condensate flashes into steam at the lower pressure and contributes toward enthalpy increase of the main condensate. This flashing reduces the amount of low-pressure extraction with some loss in cycle efficiency, but the loss is more than offset by the saving in the investment in the pump required with a contact-type heater. The cascading arrangement can be used with any number of heaters.

The expression for the actual over-all station thermal efficiency of a regenerative cycle is considerably simplified by having the turbine condition line given rather than the turbine-engine efficiency. Then, for a cycle driving an electric generator and motor drives on the auxiliaries and using contact-type heaters, the actual thermal efficiency is



(a) (b)

FIG. 3-9. Schematic of reheat-regenerative cycle.

$$\eta'_{rg} = \frac{[(h_{1'} - h_{2'}) + (1 - m'_1)(h_{2'} - h_{3'}) + (1 - m'_1 - m'_2)(h_{3'} - h_{4'})]\eta_t\eta_g - AW}{(h_1 - h_{10'})/\eta_b} \qquad (3\text{-}10)$$

where m'_1, m'_2 = actual masses of bled steam different from theoretical
η_t = turbine mechanical efficiency
η_g = generator efficiency

3-5. Reheat-Regenerative Cycle. At the present time with 1200 F temperature being seriously considered as a commercial feasibility, reheat would not be applied to a cycle with a throttle pressure less than 1,200 psi. However, for pressures higher than this value, reheat would be employed and invariably combined with a regenerative cycle as shown in Fig. 3-9, employing three heaters as an example.

The ideal thermal efficiency of this cycle is

$$\eta_{rhg} = \frac{(h_1 - h_2) + (1 - m_1)(h_3 - h_4) + (1 - m_1 - m_2)(h_4 - h_5) + (1 - m_1 - m_2 - m_3)(h_5 - h_6) - PW}{(h_1 - h_{14}) + (1 - m_1)(h_3 - h_2)} \tag{3-11}$$

With the turbine condition line as a basis the actual over-all station thermal efficiency is

$$\eta'_{rhg} = \frac{\eta_t\eta_g[(h_{1'} - h_{2'}) + (1 - m'_1)(h_{3'} - h_{4'}) + (1 - m'_1 - m'_2)(h_{4'} - h_{5'}) + (1 - m'_1 - m'_2 - m'_3)(h_{5'} - h_{6'})] - AW}{[(h_{1'} - h_{14'}) + (1 - m'_1)(h_{3'} - h_{2'})]/\eta_b} \tag{3-12}$$

In all the foregoing cycles the thermal efficiency is an absolute measure of the relative effectiveness of those cycles. However, a common designation of cycle performance is the heat rate,

$$HR = \frac{3{,}413}{\eta} \qquad \text{Btu per kwhr} \tag{3-13}$$

where η is the thermal efficiency.

The steam rate is also usually quoted to permit calculation of the boiler capacity required. The steam rate is

$$SR = \frac{\eta_b HR}{h_1 - h_f} \qquad \text{lb per kwhr} \tag{3-14}$$

where η_b = boiler or steam-generator efficiency
h_1 = initial steam enthalpy leaving steam generator, Btu per lb
h_f = feedwater enthalpy entering steam generator, Btu per lb
HR = over-all station heat rate, Btu per kwhr

3-6. Binary Vapor Cycles. Cycles using two fluids with a heat-transfer link between them have been applied to a limited extent; the best known uses water and mercury. At comparable temperatures the mercury pressure is much lower than for water. Being a liquid metal with low specific heat, mercury's saturated liquid line on the *T-S* chart is very steep, approaching the Carnot cycle. This improves actual thermal efficiency.

Figure 3-10 shows the layout of the cycle in elementary fashion. This ideal *T-S* chart assumes a temperature difference to transfer heat from the upper to the lower vapor. A mercury-steam cycle needs about 10 lb Hg for every pound of steam flow, since the latent heat of mercury is about one-tenth that of water. On the *T-S* diagram the entropy scale is drawn for 10 lb Hg and 1 lb H_2O. Mercury leaves the mercury boiler at 1 in a dry saturated state and expands isentropically in the ideal mercury turbine to state 2. It then enters the mercury-water heat exchanger (condenser-boiler) to give up its heat to the water by condensing to a saturated liquid at state 3. After passing through the mercury pump, it returns to the mercury boiler in state 4 for reevaporation to state 1.

The water leaves the condenser-boiler as a dry saturated vapor at 5 and passes through a superheater in the mercury-boiler furnace to 6. The steam enters the steam turbine, expands isentropically to 7, then condenses to a saturated liquid at 8. The pump pushes the water back to the condenser-boiler as a compressed liquid at 9.

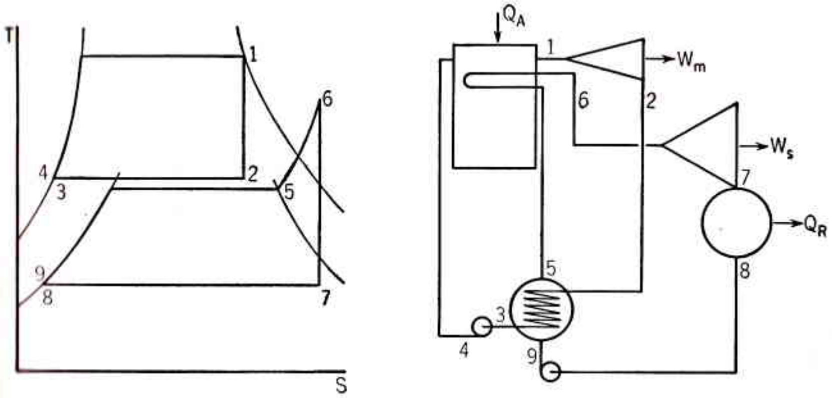


Fig. 3-10. Schematic of binary vapor cycle.

The *T-S* diagram shows the heat added to the cycle as the areas under 4-1 and 5-6; heat rejected is the area under 7-8. The total area enclosed by the two process diagrams equals the net work of the cycle. The rectangular shape of the over-all diagram indicates the basically high efficiency of the cycle. These plants have run at heat rates under 10,000 Btu per kwhr. The few plants that have been built are losing favor because of their high initial cost and maintenance cost.

GAS POWER CYCLES

3-7. Spark-ignition Cycle. In gas power cycles a fuel-air mixture burns inside a cylinder, and the combustion gases act directly as the working fluid. Beau de Rochas suggested an internal-combustion engine cycle as early as 1862. A German engineer, Otto, built a successful engine on this basis in 1876, and the so-called "Otto cycle" emerged as the accepted reference.

Figure 3-11*a* and *b* shows the theoretical cycle using air as the working fluid. A source transfers heat to the air through a conducting cylinder head to raise the air pressure; after expansion the air transfers its remaining heat to a receiver through the same cylinder head. The ideal processes are:

1-2 Addition of heat at constant volume Q_a
2-3 Isentropic expansion of the air in the cylinder transferring work to the piston
3-4 Rejection of heat at constant volume Q_r
4-1 Isentropic compression of the air by the piston

The thermal efficiency of the air-standard Otto cycle is

$$\eta_t = \frac{W}{Q_a} = \frac{Q_a - Q_r}{Q_a} = 1 - \left(\frac{V_1}{V_4}\right)^{k-1} = 1 - \frac{1}{r_v{}^{k-1}} \qquad (3\text{-}15)$$

where r_v = compression ratio = V_4/V_1
k = specific-heat ratio = c_p/c_v = 1.3 to 1.4 for air

Equation (3-15) shows that high compression will yield high efficiencies.

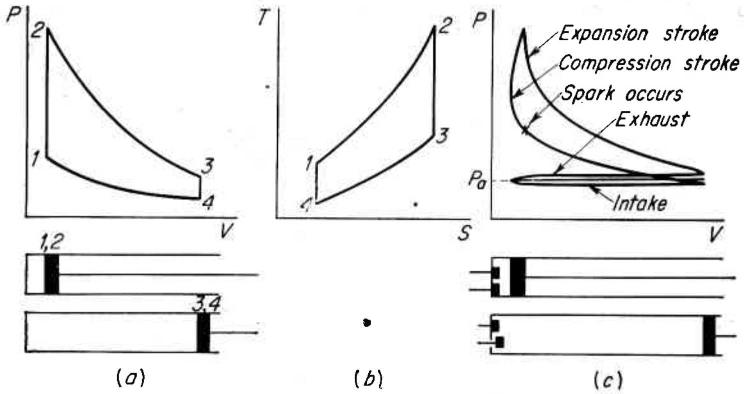


FIG. 3-11. (*a*) and (*b*) Air-standard Otto cycle; (*c*) *P-V* diagram (indicator card) of a typical s-i engine using four strokes with fuel burning in air within the engine.

Figure 3-11 compares an actual cycle with a theoretical Otto cycle. These factors influence the actual cycle and are not accounted for in the theoretical:

1. Heat transfers into or out of the gas in all processes.
2. Throttling affects intake and exhaust pressures in cylinder.
3. Actual engines mix clearance gas with fresh fuel-air mixture.
4. Dissociation of combustion gases.
5. Variation of specific heat with temperature.
6. Variation of fuel-air ratio during operation.
7. Characteristics of various fuels.
8. Ignition and valve timing.

The shaft output of an actual engine may be computed from:

$$shp = ihp \times \eta_m = \left(\rho_a \times A \times L \times N \times \frac{rpm}{n} \times \eta_v \times \frac{F}{A} \times HHV \times \eta_{ts}\right)\frac{J}{33{,}000} \qquad (3\text{-}16)$$

where shp = shaft horsepower
ihp = indicated horsepower
HHV = higher heating value of fuel, Btu per lb
ρ_a = entering air density, lb per cu ft
F/A = fuel-air ratio, lb per lb
L = piston stroke, ft
A = piston area, sq ft
rpm = engine speed, revolutions per minute
n = number of piston strokes per cycle
η_v = volumetric efficiency
N = number of cylinders
η_m = mechanical efficiency
η_{ts} = shaft thermal efficiency

A carburetor usually meters fuel into an air stream entering the spark-ignition (s-i) engine. Present development work aims at injecting fuel into the air stream ahead of the intake valve with a fuel pump. This should give better air-fuel ratio control in multicylinder engines.

3-8. Compression-ignition Cycle. A compression-ignition (c-i) engine depends on the high temperature of air compressed in a cylinder to ignite fuel separately injected into the cylinder. Figure 3-12 shows a theoretical and an actual cycle for this type of engine; the processes are:

5-1 Isentropic compression
1-2 Heat addition at constant volume Q_{av}
2-3 Heat addition at constant pressure Q_{ap}
3-4 Isentropic expansion
4-5 Heat rejection at constant volume Q_r

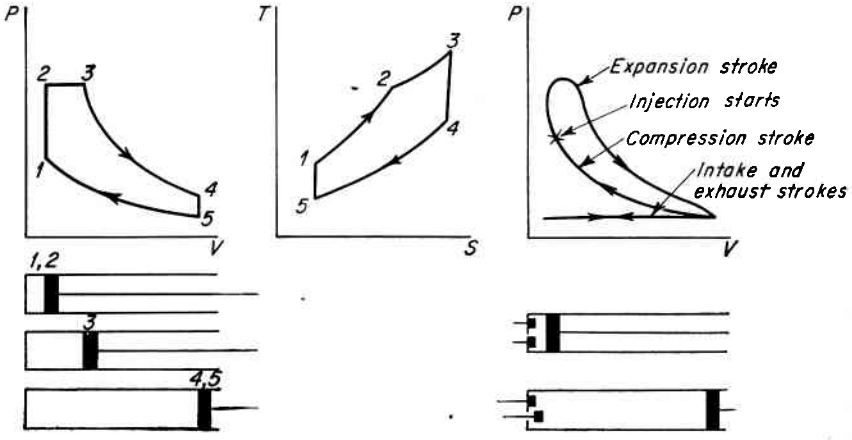
The diesel cycle has all its heat added at constant pressure; Fig. 3-12 shows the dual, or combination, cycle. The thermal efficiency of this cycle is

$$\eta_t = \frac{Q_{av} + Q_{ap} - Q_r}{Q_{av} + Q_{ap}} = 1 - \frac{1}{(V_5/V_1)^{k-1}} \frac{(P_2/P_1)(V_3/V_2)^k - 1}{[(P_2/P)_1 - 1] + k(P_2/P_1)[(V_3/V_2) - 1]} \qquad (3\text{-}17)$$

where P = pressure, psia or psfa
V = volume, cu ft per lb
k = 1.3 to 1.4 for air as standard fluid

Practical engines compress air through pressure ratios of 12 to 18 in a cylinder and then inject the fuel into the high-pressure high-temperature air. A pump forces the fuel oil through a spray nozzle in the cylinder head. Essentially the same factors as in Sec. 3-7 affect the combustion, compression, and expansion of the gases in the c-i engine. Equation (3-16) may also be used for a c-i engine to calculate the shaft horsepower.

Combustion passes through three phases in c-i engines: (1) *Ignition delay* is the time interval from the instant fuel first leaves the injection nozzle until combustion starts. (2) The second phase burns fuel admitted during the delay period; this approaches constant-volume burning with rapid pressure rise that may produce knock. (3) During the third phase fuel ignites as it enters the cylinder and burns at about constant pressure. Figure 3-12c shows these three phases in the upper shape of the cycle loop.



(a) (b) (c)

FIG. 3-12. Air-standard cycle as in (a) and (b) is present-day reference basis for c-i engines. Cycle also called dual or combination cycle. Indicator card as in (c) of actual engine is practical counterpart of (a). Note that pressure scale is so large that intake and exhaust pressure differences cannot be shown to scale.

3-9. Free-piston Compressor Cycle. Figure 3-13 shows the main elements of an engine to produce hot pressurized gas that may be used to run a gas turbine. Two free-floating pistons compress air in the inner end cylinders for delivery to a combustion cylinder between the two pistons. Linkages synchronize piston movements and fuel injection. The fuel-air mixture burns in the cylinder, and the combustion gases expand to move the two pistons outward. As the inner pistons uncover the exhaust ports, the still hot and pressurized combustion gas leaves to enter a gas turbine not shown.

The intermittent operation and combustion in the cylinder and water cooling allow the gases to reach high temperatures without needing special alloys. Efficiencies as good as with high-compression c-i engines may be realized. The exhaust, or power, gas has temperatures of about 1000 F and pressures up to 75 psia.

3-10. Simple Gas-turbine Cycle. The combustion gas turbine runs on the Brayton or Joule cycle as in Fig. 3-14, drawn as an air-standard cycle. The processes are: *1-2*, isentropic compression; *2-3*, heat addition at constant pressure Q_a; *3-4*, isentropic expansion; and *4-1*, heat rejection at constant pressure Q_r.

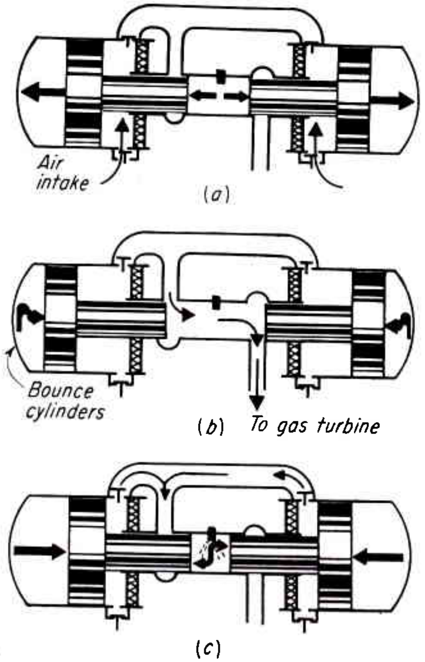


FIG. 3-13. Schematic of free-piston compressor. (a) Opposed pistons forced outward by c-i cycle compress air in bounce chamber and cause air intake; (b) exhaust ports, then scavenge ports open, and combustion gases flow to gas turbine; (c) pistons return by energy stored in bounce cylinders.

The thermal efficiency for the ideal cycle is

$$\eta_t = \frac{Q_a - Q_r}{Q_a} = 1 - \frac{1}{r_p^{(k-1)/k}} = 1 - \frac{1}{r_v^{k-1}} \tag{3-18}$$

where r_p = pressure ratio = P_2/P_1
r_v = compression ratio = V_1/V_2
k = 1.3 to 1.4 for air as standard fluid

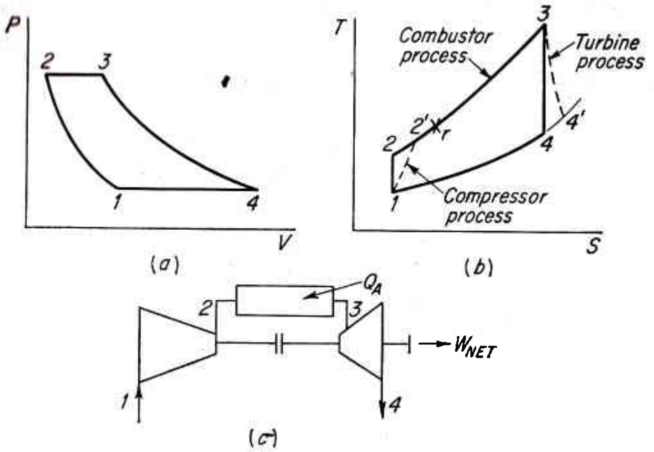


FIG. 3-14. Air-standard analysis of Brayton (Joule) cycle. Solid lines show reversible processes. Dashed lines indicate principal irreversible effects of actual cycle.

When a simple cycle is used for the actual turbine, its thermal efficiency is less than the equivalent ideal. Referring to Fig. 3-14*b*, the over-all thermal efficiency is

$$\eta_t = \frac{(h_3 - h_4)\eta_e - (h_2 - h_1)(1/\eta_c)}{(h_3 - h_2')/\eta_{co}} \tag{3-19}$$

where h = enthalpies, Btu per lb
η_e = over-all isentropic turbine engine efficiency, fraction
η_c = over-all isentropic compression efficiency, fraction
η_{co} = combustion efficiency

Actual over-all shaft thermal efficiency can also be related to other variables,

$$shp = \frac{W_A \times \dfrac{F}{A} \times HHV \times \eta_t \times J}{33{,}000} \tag{3-20}$$

where W_A is the actual air flow to the turbine unit in pounds per minute.

Figure 3-15 shows the effects of pressure ratio and combustion temperatures on gas-turbine performance. Most gas turbines work on the *simple open cycle:* atmospheric air enters the compressor, and the turbine exhausts to atmosphere. The simple cycle may also be used in a *closed* arrangement. A given mass of gas, usually air, circulates through compressor, turbine, and two heat exchangers for the heat-addition and -rejection processes.

3-11. Regenerative Gas-turbine Cycle. At any given pressure ratio the simple-cycle gas-turbine efficiency depends on gas-inlet temperature and the machine efficiencies. These act to produce a maximum efficiency, but *regeneration* under certain conditions can raise efficiency further.

A gas-to-air heat exchanger, called a "regenerator," transfers heat from turbine exhaust to compressed air entering the combustor. Most regenerators use a shell-and-tube design, though flat-plate and rotary regenerative heaters as used for boiler air heaters are being developed.

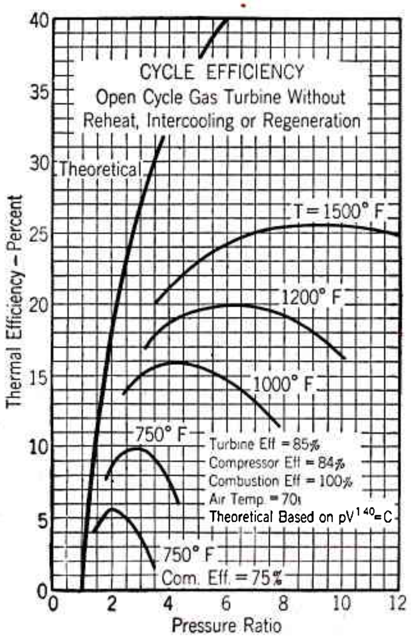


FIG. 3-15. Effect of variables on simple gas-turbine (Brayton, Joule) cycle for ideal and actual turbines. Values for actual cycles based on efficiencies shown in chart legend.

Figure 3-16 shows a gas turbine with a regenerator and calculated performances. Efficiency gain depends on both inlet-pressure and cycle-pressure ratios. Regenerator performance is measured by its *effectiveness*,

$$\eta_r = \frac{\text{actual air temperature rise}}{\text{maximum possible air temperature rise}} \tag{3-21}$$

Effectiveness varies from 0.5 to 0.8 with tolerable pressure drops in the cycle. Pressure losses must be carefully controlled because they can offset the efficiency gains of regeneration.

The thermal efficiency of the cycle in Fig. 3-16 is figured as

$$\eta_t = \frac{(h_4 - h_5)\eta_e - (h_2 - h_1)/\eta_c}{(h_4 - h_3)/\eta_{co}} \tag{3-22}$$

Figure 3-17 shows the effect of turbine gas-inlet temperature, pressure ratio, regenerator effectiveness, and compressor air-inlet temperature on cycle thermal efficiency. The air-inlet temperature also has a large effect on the unit capacity.

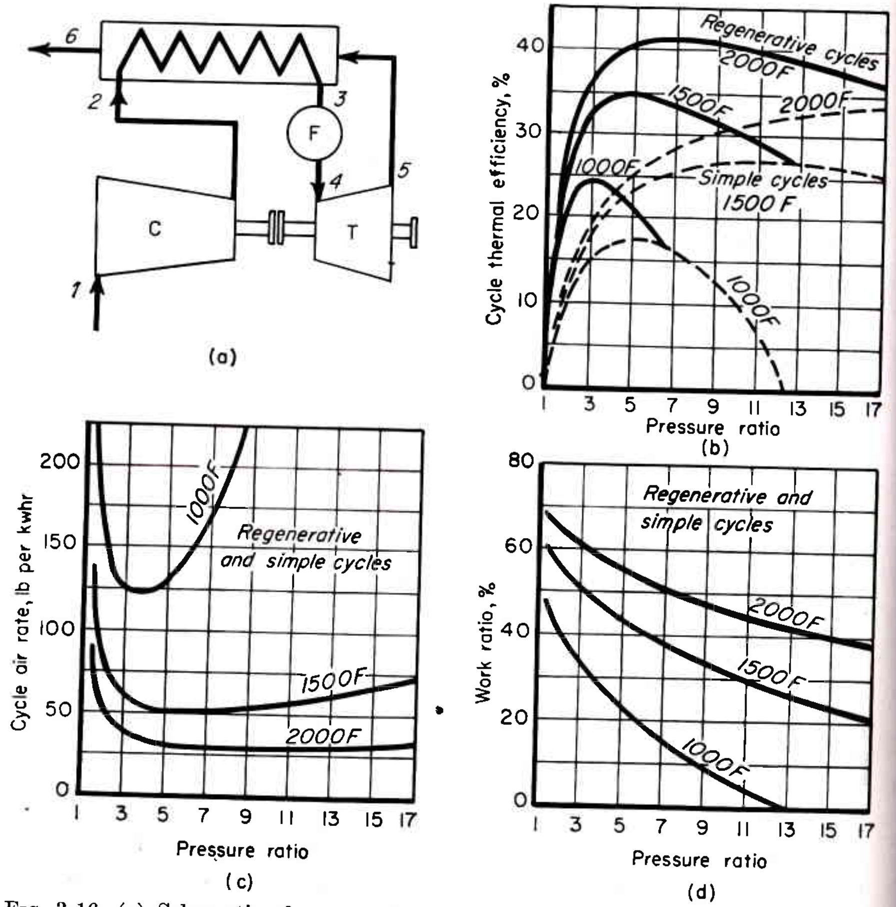


FIG. 3-16. (*a*) Schematic of regenerative gas turbine; (*b*), (*c*), and (*d*) performance based on $\eta_r = 70$ per cent regenerator effectiveness, $\eta_c = \eta_e = 85$ per cent, air inlet 60 F, $\eta_{co} = 100$ per cent in combustor *F*, no pressure losses.

3-12. Intercooling and Reheat in Gas-turbine Cycle. *Intercooling* air during compression can reduce the shaft work input to the compressor. This reflects the basic process of isothermal compression requiring minimum work. Figure 3-18 shows one- and two-stage intercooling. At higher pressure ratios intercooling raises thermal efficiency slightly. Prime benefits lie in reducing the air rate and raising the work ratio, where the latter is the ratio of cycle shaft output to total shaft work developed by the turbine. These factors are reflected in reducing the physical size of the machine for a given output.

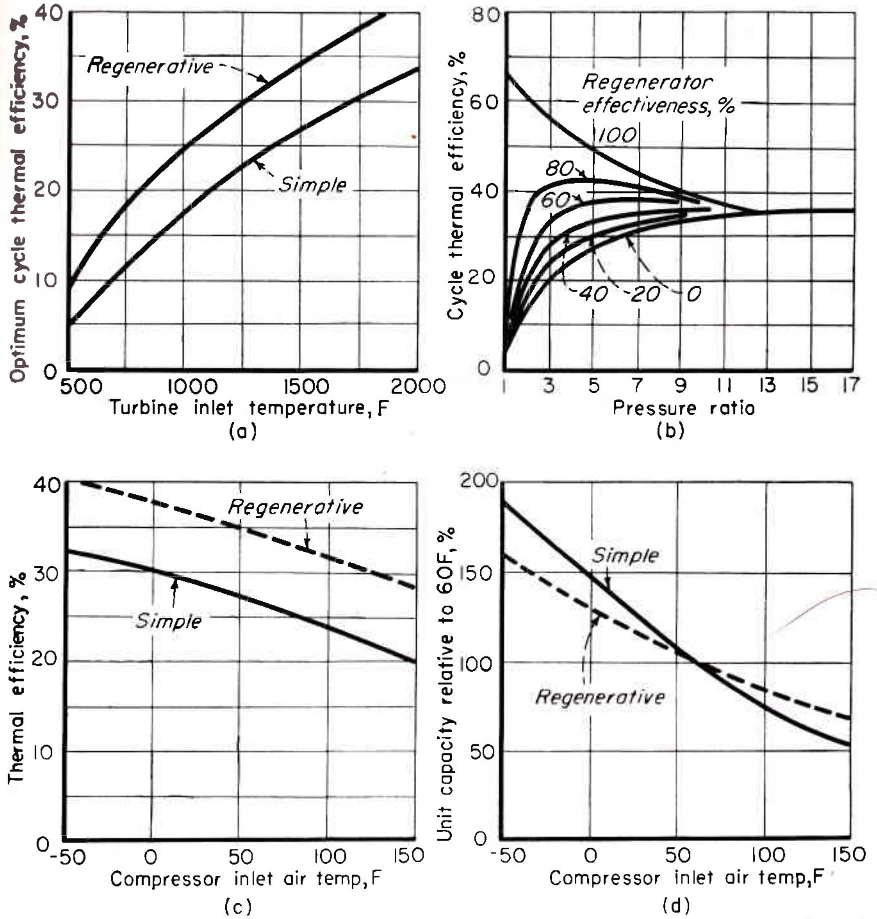


FIG. 3-17. (*a*) Effect of turbine-inlet temperature on cycle performance. (*b*) Effect of regenerator effectiveness. (*c*) Effect of compressor-inlet air temperature. (*d*) Effect of inlet-air temperature on turbine cycle capacity. Based on $\eta_e = \eta_c = 85$ per cent, $\eta_r = 70$ per cent, 1500-F inlet-gas temperature.

Equation (3-1) and Fig. 3-1 show that maximum thermal efficiency can be obtained when we have isothermal heat addition to the cycle. This proves impractical in actual cycles, but we can approach its benefits by reheating the gas after partial expansion in the turbines (Fig. 3-19). This proves helpful at higher pressure ratios, but again the prime benefit is the improvement in air rate and work ratio.

3-13. Combination Gas-turbine Cycles. We can gain maximum benefits by combining regeneration, intercooling, and reheating in gas-turbine cycles as in Fig. 3-20. We see that the thermal efficiencies for the cycle without the regenerator increase noticeably at the higher pressure ratios by adding simultaneously intercooling and reheating. The regenerator added as the third factor shows the biggest gain. These curves apply to 1500-F turbine-inlet temperature, 60-F compressor-inlet temperature, 85 per cent machine efficiencies, and a 70 per cent regenerator effectiveness. Figure 3-20*c* shows the air-rate improvement with simultaneous addition of intercooling and reheating. A regenerator, theoretically, has no influence on air rate.

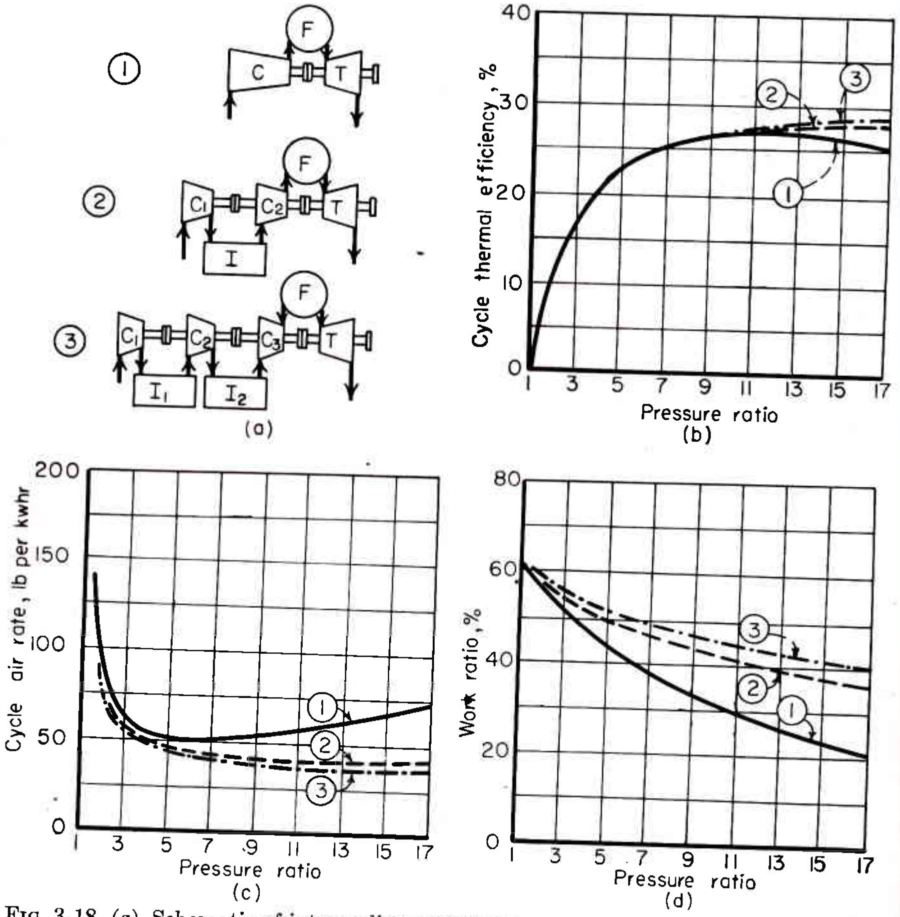


FIG. 3-18. (*a*) Schematic of intercooling applied to gas turbine; (*b*), (*c*), and (*d*) intercooled cycle performance for 1500-F inlet-gas temperature; $\eta_c = \eta_e = 85$ per cent, inlet air 60 F, $\eta_{co} = 100$ per cent, no pressure losses.

Figure 3-20*d* shows that simultaneous intercooling and reheating also improve the work ratio. Again the regenerator theoretically has little effect on this performance factor.

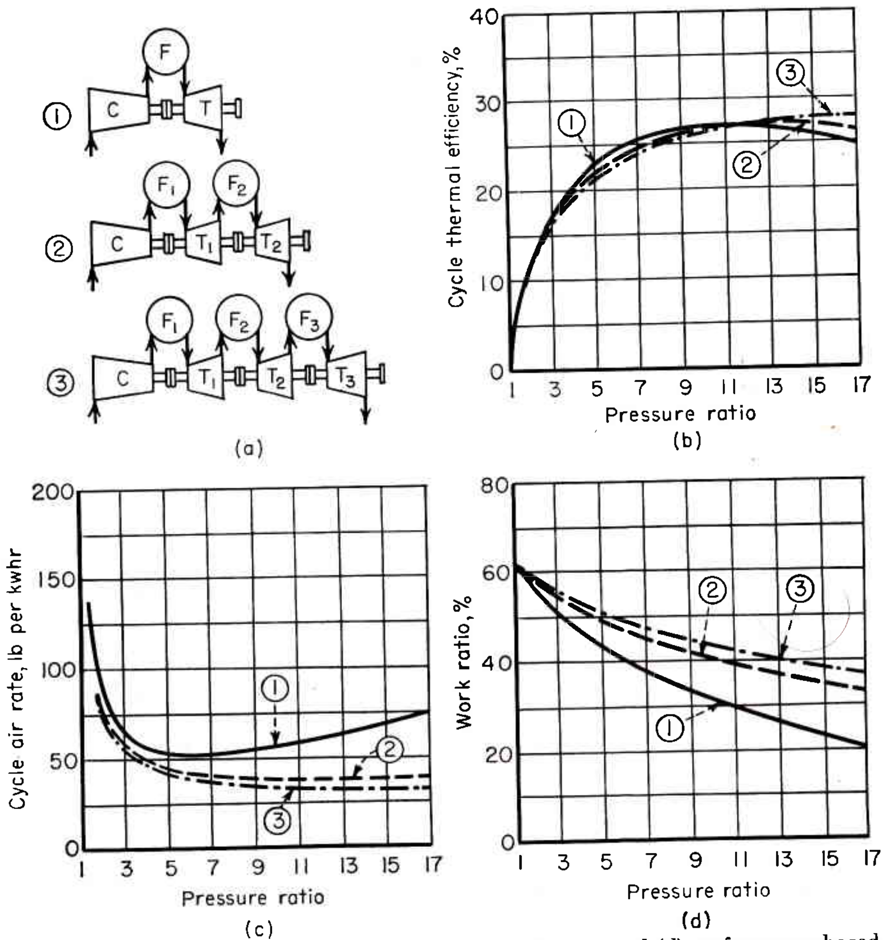
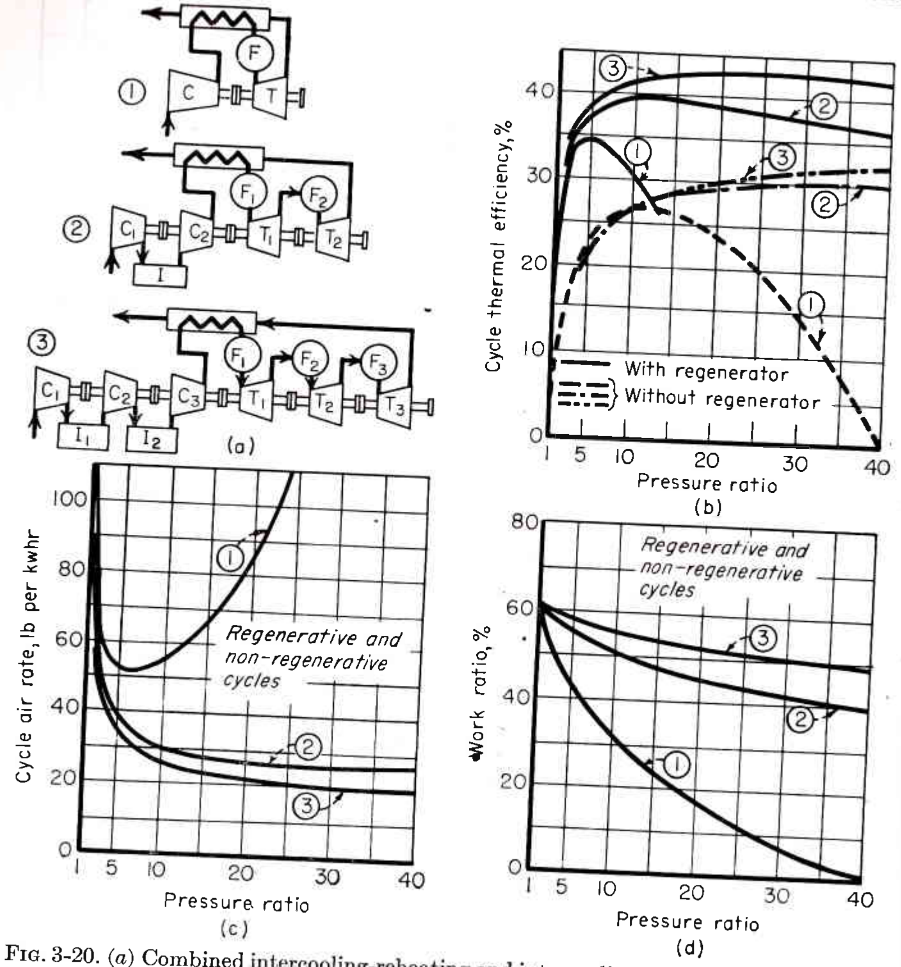


FIG. 3-19. (*a*) Schematic of reheat in gas turbine; (*b*), (*c*), and (*d*) performance based on 1500-F inlet gas; $\eta_e = \eta_c = 85$ per cent, air inlet 60 F, $\eta_{co} = 100$ per cent, no pressure losses.

A wide choice of cycle arrangement can be made for gas-turbine plants. The gain in thermal efficiency must be balanced against the cost of the additional equipment in choosing a gas-turbine plant for a given application.

3-14. Closed Gas-turbine Cycles. Figure 3-21 shows a *simple-closed-cycle* gas-turbine plant. Figures 3-16 to 3-20 show that pressure ratio, and not absolute pressures, governs the cycle performance. By closing the cycle so that that turbine exhaust enters the compressor inlet (through a recooler) we can raise the low-pressure end of the cycle to any level, and for any set of conditions we can choose the best pressure ratio. The advantages of doing this are: (1) high-density cycle fluid needs smaller volumes and hence smaller machines, (2) clean cycle fluid reduces fouling deposits on turbine blades, (3) thermal efficiency can be kept high at part loads, (4) there is better heat transfer in the air heater and recooler, and (5) gases with better thermodynamic properties than air can be used.



(a)

(b)

(c)

(d)

FIG. 3-20. (a) Combined intercooling-reheating and intercooling-reheating-regenerative cycle performance for 1500-F inlet temperature, $\eta_e = \eta_c = 85$ per cent, $\eta_r = 70$ per cent, air inlet 60 F, $\eta_{co} = 100$ per cent; (b) thermal efficiency; (c) air rate; (d) work ratio.

The closed cycle gains these advantages at the cost of: (1) adding a recooler, (2) adding an air heater, (3) using an external combustion system with more auxiliaries, and (4) using a more complex over-all control system.

3-15. Water Injection. Some improvement in efficiency or capacity may be made by injecting water into the cycle at the compressor inlet, compressor discharge, or combustor discharge. This increases the mass of fluid in the cycle to increase output but may not always raise the over-all cycle thermal efficiency.

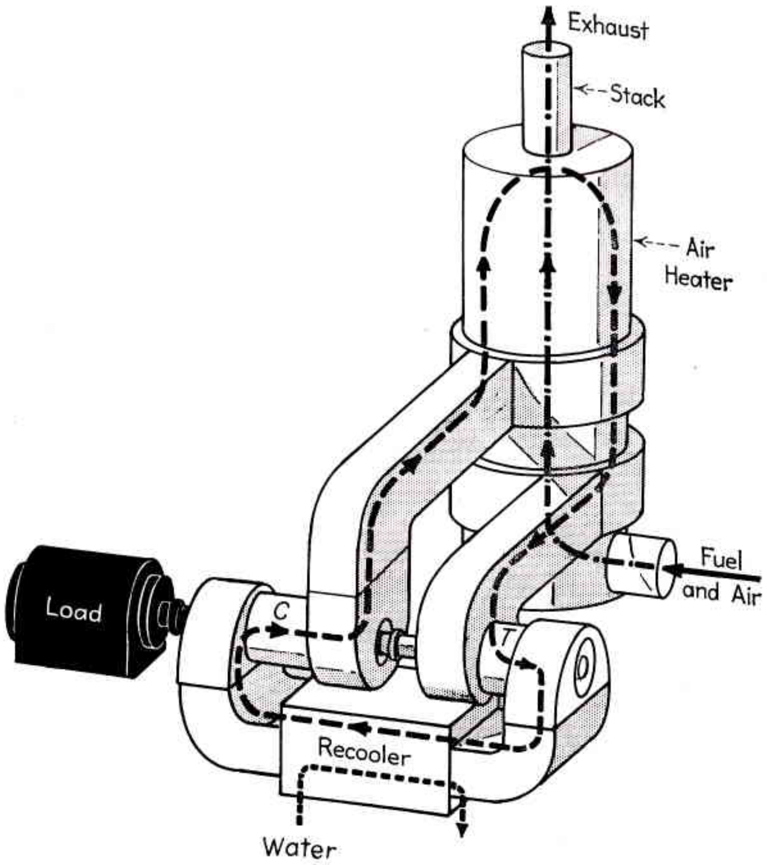


FIG. 3-21. Simple closed cycle circulates air (or other gas) only, and firing is indirect.

Some thermodynamic studies for specific cycle conditions show that gas-turbine cycles can be improved in efficiency about 1.5 per cent and in capacity by about 40 per cent by injecting water into the air leaving the compressor and before entering the combustor.

PROBLEMS

3-1. Air is heated at constant volume in a cycle from 100 F initially by the addition of 180 Btu per lb. Determine (*a*) the available energy, (*b*) thermal efficiency referred to a 60-F receiver temperature.

3-2. A Carnot-type cycle using steam operates between 500 and 80 F. Heat input changes saturated liquid to dry steam. Machine efficiencies of isentropic processes are equal. Determine (*a*) machine efficiency for a net work output equal to zero; (*b*) machine efficiency if the actual thermal efficiency is half the ideal.

3-3. Steam at 1,000 psia and 740 F is used in a Rankine nonextraction type of cycle. A pressure drop of 10 per cent occurs in the turbine valves. Steam expands to 1 in. Hg abs. The turbine engine efficiency is 85 per cent, and the feedwater-pump compression efficiency is 68 per cent. Determine (*a*) cycle thermal efficiency and (*b*) over-all station efficiency based on a steam-generator efficiency of 88 per cent; (*c*) station heat rate.

3-4. Check the theoretical efficiency at 800 F and 5,500 psia of a superpressure Rankine cycle as in Fig. 3-4.

3-5. Check the theoretical efficiency of a reheat cycle at 2,000 psia and 800 F as in Fig. 3-6 for optimum conditions.

3-6. A single reheat cycle at 500 psia with throttle and reheat temperatures at 800 F and condenser pressure 0.5 in. Hg abs operates at optimum theoretical reheat pressure. Actual cycle thermal efficiency is 20 per cent less than ideal. Feedwater-pump efficiency is 70 per cent and boiler efficiency 87 per cent. Determine the engine efficiency of each section of the turbine if the first section is expected to have a machine efficiency 5 per cent greater than the second.

3-7. An ideal three-stage regenerative cycle uses steam at 2,000 psia and 1050 F with 0.5 in. Hg back pressure. Determine the cycle thermal efficiency if heaters are most effectively located.

3-8. A two-stage regenerative cycle uses steam at 1,000 psia and 1000 F with 1.0 in. Hg back pressure. Station auxiliary power is 15 per cent of turbine output. The over-all electrical and mechanical efficiency of the turbine generator is 85 per cent, and the boiler efficiency is 86 per cent. The turbine has an engine efficiency of 88 per cent. Compute the actual over-all station heat rate.

3-9. A plant having five 20,000-kw units using steam at 200 psig and 500 F has a full-load water rate of 13.0 lb per kwhr. The steaming capacity of the existing 25 boilers exactly matches the turbine requirements. If this plant were topped by a 10,000-kw turbine having an engine efficiency of 77.6 per cent and a mechanical efficiency of 99 per cent and if the generator efficiency is 98 per cent, find:

a. The initial steam temperature required for 1,000 psi initial steam pressure.

b. The steam exhausted from the topping turbine.

c. The number of existing low-pressure boilers that must be retained to develop the full capacity of the plant.

d. Compare the steam outputs of the original and future boiler rooms.

e. Compare the kilowatt capacity of the original and the future plants.

3-10. In an electric generating plant using a binary vapor cycle with mercury in the upper cycle and steam in the lower the ratio of mercury flow to steam flow is 10 : 1 on a mass basis. At an evaporation rate of 2,000,000 lb per hr for the mercury its specific enthalpy rises by 150 Btu in passing through the boiler. Superheating the steam in the boiler furnace adds 250 Btu to the steam specific enthalpy. The mercury gives up 110 Btu per lb during condensation, and the steam gives up 900 Btu per lb in its condenser. Radiation losses are negligible. The over-all boiler efficiency is 85 per cent. The combined turbine mechanical and generator efficiencies are each 95 per cent for the mercury and steam units. The station auxiliaries require 5 per cent of the energy generated by the units. Find the over-all heat rate of the plant in Btu per kilowatthour.

3-11. A s-i engine has the following specifications and characteristics when operating at full throttle:

Shaft output, 100 hp
Six cylinders, 4 in. × 4½ in.
Clearance volume, 10.2 cu in.
Speed, 2,800 rpm

Air inlet, 60 F
Fuel, average gasoline
Fuel rate, 56.2 lb per hr
Air rate, 644.8 lb per hr

Determine the ratio of actual to ideal (air standard, $k = 1.3$) thermal efficiency.

3-12. Derive Eq. (3-17) for thermal efficiency of a dual cycle on an air-standard basis.

3-13. Air at 60 F is used in an ideal air-standard open gas-turbine cycle with a pressure ratio of 6 and maximum temperature of 1400 F. Determine (*a*) the effect on thermal efficiency if machine efficiencies of 85 per cent and a combustion efficiency of 90 per cent are anticipated; (*b*) the work ratio of the ideal and actual cycle.

3-14. A gas-turbine unit operates with a pressure ratio of 4, machine efficiencies of 86 per cent, and gas temperature of 1200 F. Compute (*a*) thermal efficiency; (*b*) work ratio.

3-15. Assume that a regenerator with 70 per cent regenerator effectiveness is added to the unit of Prob. 3-14. Compute (*a*) thermal efficiency; (*b*) work ratio.

3-16. A nonregenerative gas-turbine unit operates with a pressure ratio of 6 and machine efficiencies of 84 per cent, with a maximum temperature of 1400 F and initial temperature of 60 F. Assume both an open cycle and a closed cycle with a low side pressure of 100 psia. Compare (*a*) thermal efficiencies; (*b*) work ratios.

3-17. Assume a regenerator with 75 per cent regenerator effectiveness added to each of the cycles in Prob. 3-16. Determine the effect on (*a*) thermal efficiency; (*b*) work ratio.

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CHAPTER 4

HEAT TRANSFER AND FLUID FRICTION

4-1. Basic Concepts. In thermodynamics we must know how much energy (quantity) there is and what the temperature variation (temperature pattern) is in a heat flow through a body. Heat-transfer rates depend on several factors; when they are known, we can predict the performance of heat-exchanging equipment. In this chapter we can only highlight the problems and rather complicated methods of studying heat transfer. Often we can only roughly estimate equipment performance.

A heat transfer *transports* energy between two bodies under the influence of a *temperature gradient* (temperature difference) between them. In *steady-state* heat transfer the temperature distribution between the bodies does not change with time. In an *unsteady state* the temperature gradient changes with time. Practical heat transfer in power plants during operation approaches steady-state conditions; this can be handled by one-dimensional equations, which will be emphasized in this chapter.

4-2. Heat-transfer Factors. *Conduction* and *radiation* are basic modes of heat transfer. Conduction transfers heat by molecular interaction through bodies in contact. Radiation transfers heat between bodies by energy propagating electromagnetic waves. *Convection*, often termed a heat-transfer method, is not a basic mode. It transfers energy by conduction or radiation or both and a concurrent mixing type of mass flow. Density changes move fluids in a *natural*, or *free*, convection as temperatures change. *Forced* convection implies mechanically induced flow by a pump or fan. Practical equipment ordinarily uses more than one method of heat transfer.

In heat transfer we must consider: (1) sources and receivers of heat flow; (2) path of heat flow. Sources may be cooling, condensing, or freezing fluids and chemical or nuclear reactions; receivers may be boiling, heating, or melting substances and chemical reactions. The path must be described in terms of area, length, thermal resistance, and temperature distribution or pattern. Electrical circuits offer helpful analogies in analyzing heat flows. Heat-transfer rates depend on some function of temperature difference and inversely on thermal resistance.

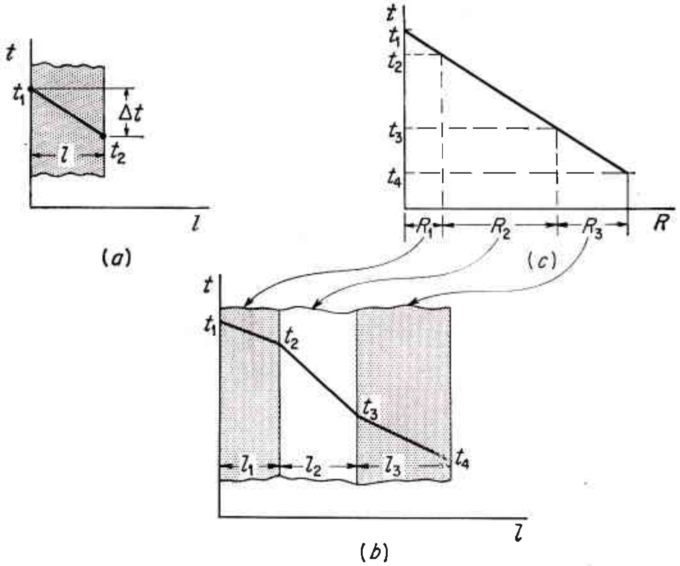


FIG. 4-1. (*a*) Temperature gradient with length for single wall; t_1 and t_2 are at wall surfaces, walls are homogeneous, specific heats and conductivities of material constant throughout. (*b*) Temperature gradient for series conduction; t_2 and t_3 depend on conductivities and wall thickness. (*c*) Temperature-resistance diagram applied to series conduction; note straight-line characteristic and consequent path temperatures when resistances are known.

4-3. Steady-state Conduction. The basic conduction equation of Fourier applied to a plane wall (Fig. 4-1*a*) is

$$q = -kA\frac{dt}{dl} \tag{4-1}$$

where q = heat transfer, Btu per hr
k = thermal conductivity, Btu per sq ft-hr-°F-ft
A = area normal to heat flow, sq ft
dt/dl = temperature gradient, °F per ft (t decreases as l increases, and so minus sign gives plus value for q)

When dt/dl is a constant negative value as in Fig. 4-1*a*,

$$q = kA\frac{\Delta t}{l} \tag{4-2}$$

which can be written

$$\frac{q}{A} = \frac{t}{l/k} = \frac{t}{R} \tag{4-3}$$

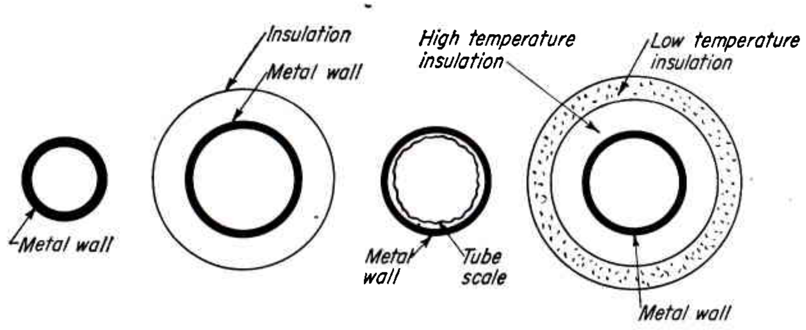
where q/A = thermal flow, Btu per hr-sq ft
R = l/k = thermal resistance, sq ft-hr-°F per Btu
l = wall thickness, ft

For composite walls or thermal resistances in series as in Fig. 4-1*b*,

$$\frac{q}{A} = \frac{k_1(t_1 - t_2)}{l_1} = \frac{k_2(t_2 - t_3)}{l_2} = \frac{k_3(t_3 - t_4)}{l_3}$$
$$= \frac{t_1 - t_4}{l_1/k_1 + l_2/k_2 + l_3/k_3} = \frac{t_1 - t_4}{\Sigma R} = \alpha(t_1 - t_4) \qquad (4\text{-}4)$$

where α is the coefficient of heat transfer for the composite wall. Value of α may be computed from individual wall thicknesses and conductivities; specific values will be found in references. Equations (4-3) and (4-4) may be used for curved walls with large radii.

Figure 4-1*c* shows a resistance diagram for series conduction flow; note the straight-line character.

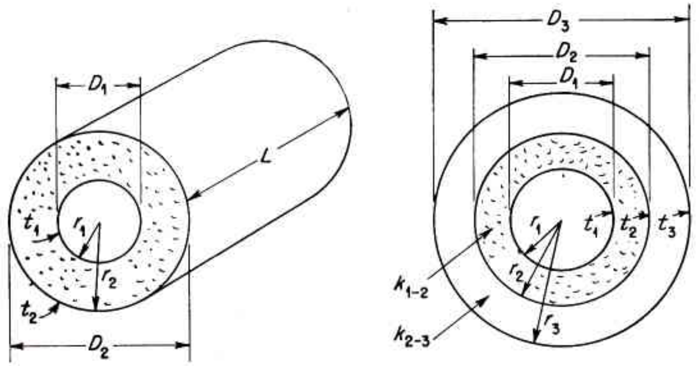


(Condenser) (Steam line) (Boiler tube) (High temperature line)

FIG. 4-2. Actual concentric wall effects in heat transfer; equipment cross sections well known in some cases, approximations (i.e., boiler scale inside tubes) in others.

4-4. Thermal Conductivities. Thermal conductivities are properties of materials and vary widely for different substances, high for metals and low for insulating substances. For any material k depends on the temperature (see Appendix). Similar materials may differ widely in conductivities. While heat does not flow through fluids by pure conduction, we use their thermal conductivities to calculate convection heat-transfer rates. Since k varies with temperature, the true temperature gradient through a material is not linear, but in practice we largely use mean values of k based on average wall temperatures. This gives us the same over-all Δt, but the internal temperature pattern (Fig. 4-1*c*) is not true when based on this k.

4-5. Conduction through Cylindrical Walls. Power equipment mostly uses tubes as heat-transfer surfaces. These may be considered as single or concentric cylinders from a heat-transfer standpoint. Figure 4-2 shows some examples that may be treated as in Fig. 4-3. For a single-thickness tube



(*a*) (*b*)

FIG. 4-3. Cylindrical sections for heat transfer; the heat-flow path is of constant length but increasing area in direction of flow. (*a*) Single wall; (*b*) multiple walls.

$$q = -kA\,\frac{dt}{dr}$$
$$dt = \frac{-q\,dr}{2\pi Lkr}$$

For constant k by integration

$$q = \frac{2\pi Lk}{\ln (r_2/r_1)}\,(t_1 - t_2) \qquad (4\text{-}5)$$

The thermal current is q/A and may be referred to either the inside or the outside tube area. Based on the inside area,

$$\frac{q}{A_1} = \frac{q}{2\pi r_1 L} = \frac{k}{r_1 \ln (r_2/r_1)}\,\Delta t \qquad (4\text{-}6)$$

Sometimes heat flow through cylinders is based on a flat-plate equivalent. For example, an area called the logarithmic mean area A_m can be computed having the same thickness, same k, and same over-all Δt as the tube and giving the same heat flow q; based on Eq. (4-2),

$$A_m = \frac{q(r_2 - r_1)}{k(t_1 - t_2)} \qquad (4\text{-}7)$$

Substituting for q from Eq. (4-5),

$$A_m = \frac{2\pi L(r_2 - r_1)}{\ln (r_2/r_1)} \qquad (4\text{-}8)$$

From this a logarithmic mean radius may be found,

$$r_m = \frac{A_m}{2\pi L} = \frac{r_2 - r_1}{\ln (r_2/r_1)} \qquad (4\text{-}9)$$

For thick-walled cylinders use Eq. (4-7) to solve for q and Eq. (4-8) to find A_m. Thin-walled cylinders ($r_1 > 0.7r_2$) can be calculated by using the arithmetic mean area. For single circular sections the thermal resistance is

$$R = \frac{r_1 \ln (r_2/r_1)}{k} \tag{4-10}$$

when based on inner area A_1.

For multiple walls (Fig. 4-3*b*) the unit heat flow is

$$\frac{q}{A_1} = \frac{t_1 - t_3}{\Sigma R} = \frac{t_1 - t_3}{r_1 \left[\dfrac{\ln (r_2/r_1)}{k_{1-2}} + \dfrac{\ln (r_3/r_2)}{k_{2-3}} \right]} \tag{4-11}$$

where k_{1-2} and k_{2-3} are thermal conductivities in Btu per square foot per hour per degree Fahrenheit per foot for the successive wall materials.

4-6. Combination Heat-transfer Effects. In power plants heat flows by conduction through refractory furnace walls, boiler and waterwall tubes, feedwater heater tubes, condenser tubes, etc. In all this equipment conduction is only part of the total effect. Figure 4-4 shows a frequent heat-transfer arrangement: from a source fluid, through a wall, to a receiver fluid. Usually the fluid heat transfer depends more on the fluid films clinging to the wall surfaces than the wall itself. The film conductivities are much lower than the metal-wall conductivity. These films are very thin and stationary, while the fluids flow at considerable velocity,

$$\frac{q}{A} = h_1(t_s - t_{w_1}) = \alpha(t_{w_1} - t_{w_2}) = h_2(t_{w_2} - t_r) \tag{4-12}$$

where h_1, h_2 = film coefficients of heat transfer for convection, Btu per sq ft-hr-°F
α = wall coefficient = k/l for a single material plane wall

Also

$$\frac{q}{A} = \frac{t_s - t_r}{\Sigma R} = U(t_s - t_r) \tag{4-13}$$

$$\Sigma R = \frac{1}{h_1} + \frac{1}{\alpha} + \frac{1}{h_2} = \frac{1}{U} \tag{4-14}$$

where U is the over-all heat-transfer coefficient in Btu per square foot per hour per degree Fahrenheit. Film effects must be accounted for also on circular or curved walls of tubes. For radial-heat-flow paths as in Fig. 4-3*a* the over-all resistance based on the inner wall is

$$\Sigma R = \frac{1}{h_1} + \frac{1}{h_2} + \frac{r_1 \ln (r_2/r_1)}{k} = \frac{1}{U} \tag{4-15}$$

where h_1, h_2 = film coefficients based on inner-wall area
U = over-all coefficient of heat transfer based on inner-wall area

In all conduction through circular walls the inner wall has been arbitrarily taken as the reference area; the outer wall may also be used. In using a consistent area for circular walls the temperature-resistance diagrams (Fig. 4-4*b*) can be applied just as with a plane wall.

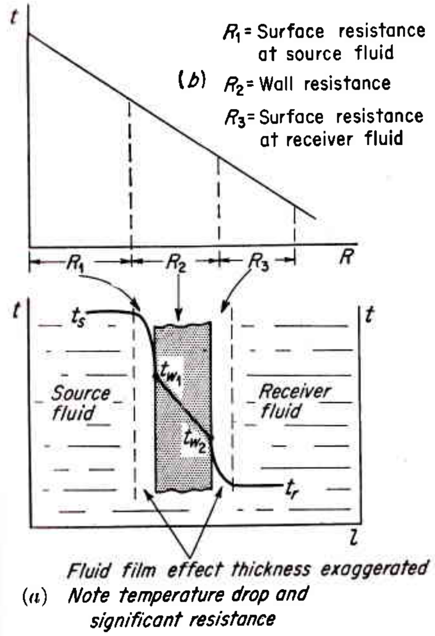

t
R_1 = Surface resistance at source fluid
(b) R_2 = Wall resistance
R_3 = Surface resistance at receiver fluid
R_1
R_2
R_3
R
t
t_s
t_{w_1}
Source fluid
Receiver fluid
t_{w_2}
t_r
t
l
Fluid film effect thickness exaggerated
(a) Note temperature drop and significant resistance

FIG. 4-4. (*a*) Surface-film effects as factors in over-all heat transfer; note that over-all temperature drop is from source fluid t_s to receiver fluid t_r. (*b*) Temperature-resistance diagram applies also to film effects; note that $t_{w_1} - t_{w_2}$ for thin metal walls may be insignificant compared with $t_s - t_{w_1}$ or $t_{w_2} - t_r$.

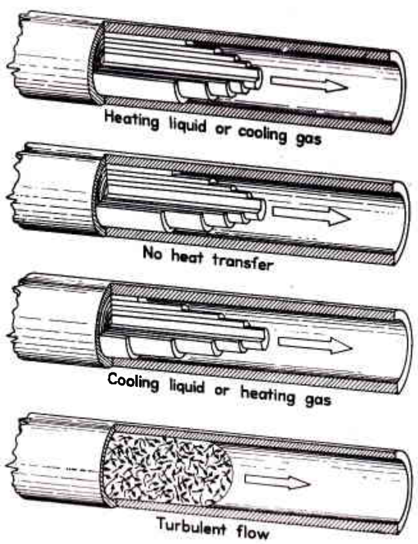


FIG. 4-5. Fluid flow may be streamline or turbulent. With streamline flow, no heat transfer, a parabolic velocity profile exists. Heat transfer changes velocity distribution as in center two sketches. Turbulent flow means eddies and crosscurrents with relatively flat velocity profile, lower sketch.

4-7. Flow Patterns in Tubes. Fluids forced through tubes may flow in *streamline* or *turbulent* fashion. The flow downstream from the entrance of circular tubes will be streamline if the Reynolds number, N_{Re} is less than about 2,100. The flow will be laminar and the velocity profile parabolic in shape. Fluids with N_{Re} greater than about 4,000 flow turbulently and have less velocity variation across the section, a fairly flat velocity profile. Flow patterns for N_{Re} between 2,100 and 4,000 are a blend of the two basic types of flow.

Flow of either type may take place with heat transfer (Fig. 4-5).

Streamline flow has no cross velocities, and heat transfers by conduction in the fluid. In turbulent flow the eddies with their cross velocities greatly aid heat transfer in addition to the conduction. Radiation effects step up heat transfer in gases. Turbulent flow has higher heat-transfer coefficients.

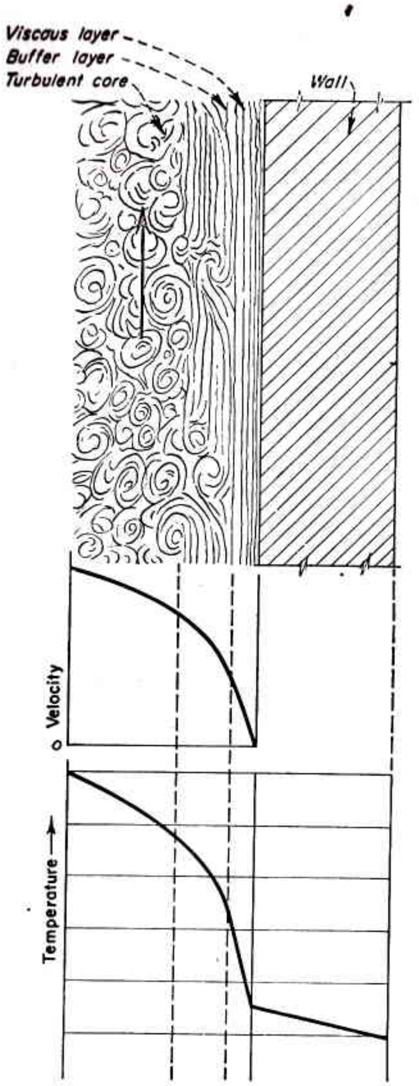


FIG. 4-6. Flow pattern close to wall in turbulent flow. Heat-transfer resistance found mainly in laminar layer across which significant temperature differences and flow velocities occur.

In turbulent flow boundary layers of fluid separate the main stream and the wall (Fig. 4-6). These layers have high heat-flow resistance and create large temperature gradients (Fig. 4-4). Many factors affect the heat-transfer characteristics of these boundary films. Successful heat-transfer solutions depend on knowing the coefficients for these films and how they vary with convection conditions.

4-8. Convection Heat-transfer Coefficients. These coefficients depend on: (1) type and rate of fluid motion; (2) properties of the fluid; (3) size of the fluid-flow path; and (4) nature of state change, if any, produced by heat transfer. We shall consider the latter in a later article. Power-plant processes have no functional relation by which the coefficient can be simply evaluated for all cases. We use dimensional analysis to set up groups of variables that correlate with film coefficients. Plotting experimental data leads to equations that can be used for specific convection-flow conditions.

Fluid properties enter into these equations, and usually all properties vary with temperature. Temperature gradients in the boundary film vary as the flow progresses over the heat-transfer surface. Some form of averaging must be used to simplify calculations. The value giving

best correlation from experimental data usually is a temperature corresponding to the average fluid properties of the boundary film.

Table 4-1 lists some common fluid properties and flows used in later applications.

TABLE 4-1. COMMON FLUID PROPERTIES AND FLOWS

Variable	Dimensions	Units
Mass	M	lb_m
Length L or diameter D	L	ft
Time	θ	hr
Temperature	T	°F
Energy (heat)	Q	Btu
Velocity, $\bar{V}$	l/θ	ft per hr
Viscosity, μ	ML/θ	lb_m per hr-ft
Density, ρ	M/L^3	lb_m per cu ft
Specific heat, c_p	Q/MT	Btu per lb_m-°F
Conductivity, k	$Q/L\theta T$	Btu per sq ft-hr-°F-ft
Heat-transfer coefficient, h	$Q/L^2\theta T$	Btu per sq ft-hr-°F
Mass current, G	$M/\theta L^2$	lb_m per sq ft-hr
Mass-flow rate, w	M/θ	lb_m per hr

Other specialized texts fully treat the analyses of convection heat transfer through dimensional analysis. From these certain dimensionless groups of variables are shown to be significant in convection studies (Table 4-2).

TABLE 4-2. SELECTED DIMENSIONLESS GROUPS

Group name	Symbol	Formula
Reynolds number	N_{Re}	$\frac{D\bar{V}\rho}{\mu} = \frac{DG}{\mu}$
Nusselt number	N_{Nu}	$\frac{hD}{k}$
Prandtl number	N_{Pr}	$\frac{c_p\mu}{k}$
Stanton number	N_{St}	$\frac{h}{c_p\bar{V}\rho} = \frac{h}{c_pG}$
Peclet number	N_{Pe}	$\frac{\rho\bar{V}Dc_p}{k} = N_{Re}N_{Pr}$
Grashof number	N_{Gr}	$\frac{g\beta\theta L^3\rho^2}{\mu^2}$
Group products	j	$N_{St}(N_{Pr})^{2/3}$

4-9. Selected Heat-transfer-coefficient Equations. The literature of heat transfer carries many experimental correlations of heat-transfer data for convection flow. It should be consulted for *exact design work.* Some typical equations are as follows.

a. Turbulent Forced-convection Flow inside Long Circular Tubes ($L/D > 60$). $N_{Re} > 300$ to 10,000 (depending on fluid).

$$\frac{hD}{k} = 0.023 \left(\frac{DG}{\mu}\right)^{0.8} \left(\frac{c_p\mu}{k}\right)^n \tag{4-16}$$

where $n = 0.3$ for cooling and 0.4 for heating.

Properties are evaluated at the mean fluid temperature.

Another equation correlating fluid friction and heat transfer is

$$j = \frac{h}{c_pG}\left(\frac{c_p\mu}{k}\right)^{2/3} = \frac{0.023}{(DG/\mu)^{0.2}} \tag{4-17}$$

For gases $N_{Pr} = c_p\mu/k$ is close to unity, and Eq. (4-16) may be used as

$$\frac{hD}{k} = 0.0207 \left(\frac{DG}{\mu}\right)^{0.8} \tag{4-18}$$

b. Turbulent Forced-convection Flow inside Noncircular Channels. For a circular annulus section use

$$D = D_2 - D_1 \qquad \text{in Eqs. (4-16) to (4-18)} \tag{4-19}$$

where D_1 and D_2 are inner and outer diameters of flow path in feet. For a rectangular section use

$$D = \frac{4A}{P} \qquad \text{in Eqs. (4-16) to (4-18)} \tag{4-20}$$

where A = flow-path area, sq ft
P = wetted perimeter, ft

For parallel flow outside the tube bank use

$$D = \frac{4A}{P} \qquad \text{as in Eq. (4-20)} \tag{4-21}$$

c. Turbulent Forced-convection Flow inside Coiled Tubes. Increase values from Eqs. (4-16) to (4-18) by about 20 per cent. For air a proposed equation for a multiplier is

$$\text{Multiplier} = 1 + 3.5\frac{D_i}{D_c} \tag{4-22}$$

where D_i = inside tube diameter
D_c = coil diameter

d. Streamlined Forced-convection Flow inside Tubes (Water and Oils)

$$\frac{hD}{k} = 2.53\left(\frac{wc_p}{kD}\right)^{1/3} \tag{4-23}$$

where w is the mass flow in pounds per hour.

e. Turbulent Forced-convection Flow across Nonbaffled Tube Banks with Circular Tubes. Tubes in heat exchangers may be arranged *staggered* or *in line* (Fig. 4-7).

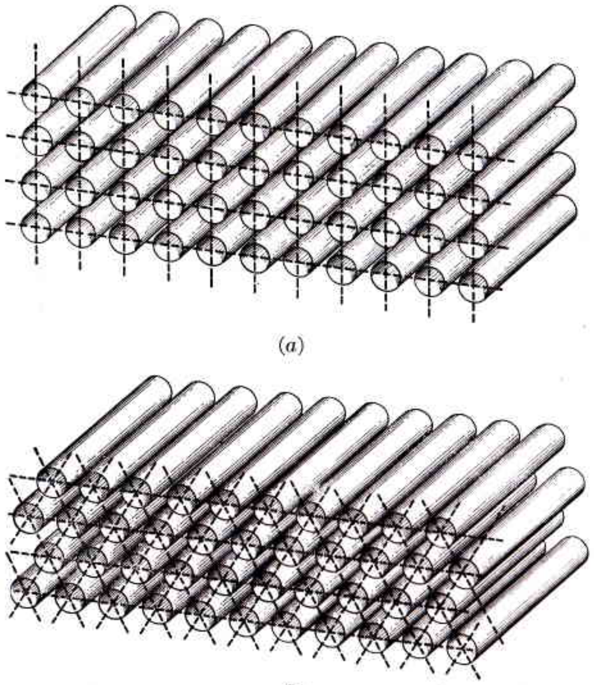


FIG. 4-7. Tubes may be arranged in either form. Staggered tubes give substantially higher coefficients of heat transfer.

1. For gas flow normal to a bank of staggered nonbaffled tubes (N_{Re} = 2,000 to 40,000; 10 rows minimum)

$$\frac{hD}{k} = 0.33\left(\frac{c_p\mu}{k}\right)^{0.33}\left(\frac{DG}{\mu}\right)^{0.6} \tag{4-24}$$

where D = outside tube diameter, ft
G = mass current based on minimum-flow area, lb$_m$ per sq ft-hr

For less than 10 rows h is multiplied by factors of the order of 0.7 for 1 tube to 1.0 for 10 tubes. For in-line tubes h is multiplied by about 0.75. Baffled tubes reduce h by a factor of about 0.6.

2. For water flow normal to a bank of staggered nonbaffled tubes ($N_{\text{Re}} > 2{,}000$)

$$h = \frac{370(1 + 0.0067t)V^{0.6}}{D^{0.4}} \tag{4-25}$$

where t = temperature, °F
V = maximum velocity, ft per sec
D = tube diameter, in.

f. Natural, or Free, Convection to Air. Free convection to air at ordinary temperatures and pressures can be estimated by simplified equations of the following forms ($X = N_{\text{Pr}}N_{\text{Gr}}$).

1. Vertical planes and cylinders:

$$h = 0.19\,\Delta t^{1/3} \qquad X = 10^9 \text{ to } 10^{12} \tag{4-26}$$

$$h = 0.29\left(\frac{\Delta t}{L}\right)^{0.25} \qquad X = 10^4 \text{ to } 10^{19} \tag{4-27}$$

2. Single horizontal cylinder:

$$h = 0.18\,\Delta t^{1/3} \qquad X = 10^9 \text{ to } 10^{12} \tag{4-28}$$

$$h = 0.27\left(\frac{\Delta t}{D}\right)^{1/3} \qquad X = 10^3 \text{ to } 10^9 \tag{4-29}$$

3. Square heated plates up or cooled plates down:

$$h = 0.22\,\Delta t^{1/3} \qquad X = 2 \times 10^7 \text{ to } 3 \times 10^{10} \tag{4-30}$$

$$h = 0.27\left(\frac{\Delta t}{L}\right)^{0.25} \qquad X = 10^5 \text{ to } 2 \times 10^7 \tag{4-31}$$

4. Square heated plates down or cooled plates up:

$$h = 0.12\left(\frac{\Delta t}{L}\right)^{0.25} \qquad X = 3 \times 10^5 \text{ to } 3 \times 10^{10} \tag{4-32}$$

4-10. Boiling Liquids and Condensing Vapors. We can compute heat transfer to liquids from tube surfaces by convection coefficients if no phase changes take place in either liquid or vapor. Boiling needs high heat-transfer rates, and additional factors come into play that affect them. Figure 4-8 shows two types of boiling for steam at atmospheric pressure.

In nucleate boiling bubbles form on the peaks of the natural surface roughness. Heat flows from the surface through the slightly superheated liquid and into the tiny steam bubbles. The bubbles grow as the heat accumulates. With increasing buoyancy the bubbles leave the heating surface to rise through the liquid and break into the vapor space above. Most equipment uses this type of boiling.

Raising the heat transfer (and temperature) too high starts film boiling. The liquid film evaporates to form a vapor film that reduces heat flow, raises metal temperatures to dangerous levels, and causes "bumping" in evaporators. The over-all coefficient may be affected more by the film resistance on the other side of the heat-transfer surface or by the scale resistance on the tubes. Boiling coefficients depend on many factors and may be of the order of 2000 to 10,000 Btu per sq ft-hr-°F.

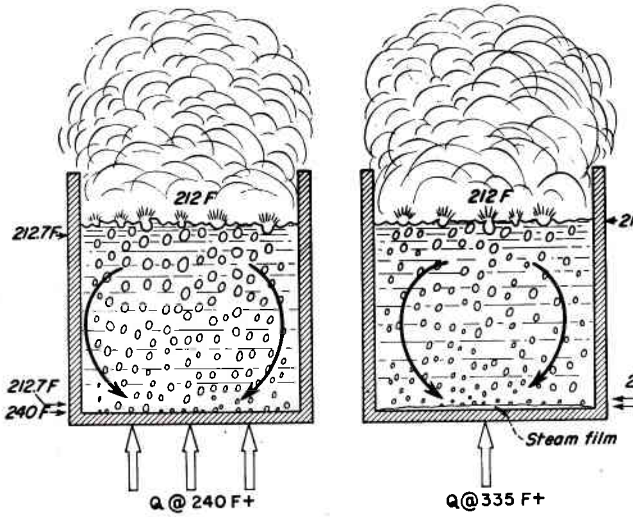
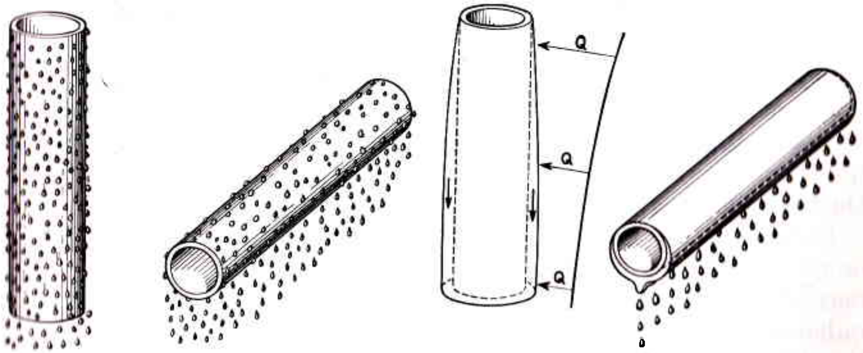


FIG. 4-8. Nucleate boiling makes best use of heat-transfer surface; too high metal temperatures form undesirable steam film.



(*a*) (*b*)

FIG. 4-9. Dropwise condensation may be made to take place for a short time under controlled conditions of polished surfaces or thin oil coatings; the filmwise type is the most common.

Figure 4-9 shows that condensation in condensers and heaters may be dropwise or filmwise. Coefficients for dropwise condensation may be about fifteen times higher than for filmwise, but in most cases the condensing surface effect is only a minor part of the over-all heat-transfer effect. Film coefficients of the receiving fluid, scale, or fouling may be the controlling factor.

Filmwise-condensation coefficients vary from 1500 to 3000 Btu per sq ft-hr-°F, and 2000 is often used in calculations. Air and other noncondensable gases reduce the coefficient, and so they should be removed by venting.

Nusselt's expression for the filmwise-condensation coefficient on clean horizontal tubes is

$$h = 103.7 \sqrt[4]{\frac{k^3 \rho^2 h_{fg}}{D\mu\, \Delta t}} \tag{4-33}$$

where h_{fg} = enthalpy of vaporization, Btu per lb
Δt = temperature difference between tube surface and vapor, °F
D = tube diameter, ft
k = thermal conductivity of liquid, Btu per hr-sq ft-°F-ft
μ = liquid viscosity, lb per ft-hr
ρ = liquid density, lb per cu ft

Modify this equation for vertical tubes by using tube length for D and increasing the h values by a factor of 1.3. The equation usually needs a trial solution. Usually a tube placed horizontally gives better heat-transfer coefficients for the condensing surface.

4-11. Radiant Heat Transfer. Radiant heat transfers energy through space without the aid of intervening media. This forms only one part of the radiant-energy spectrum that includes: cosmic rays, gamma rays, X rays, ultraviolet rays, light rays, infrared heat rays, and radio waves. Thermal radiation includes the infrared and the light with wavelengths of about 10^{-2} to 10^{-5} cm. Heat radiation underlies the operation of important power-plant equipment and depends on the temperature of the body emitting the energy.

Radiant energy travels in straight lines, may be reflected, refracted, or polarized, and follows the inverse-square law of intensity. A body may absorb, transmit, or reflect incident radiant energy. The net heat radiated between two bodies that "see" each other completely equals the difference of the emissions between each of them. Solids, liquids, and gases may radiate energy.

Water vapor and carbon dioxide are the principal sources of gaseous radiation in boiler furnaces. Consult specialized texts for a complete discussion of this topic.

A *black body* absorbs all the incident radiation striking it. This technical term does not necessarily refer to body color, though black materials usually absorb most incident energy. A perfect absorber is also a perfect radiator. The radiation spectrum from bodies includes an energy distribution that varies in intensity for the different wavelengths. The distribution varies with temperature of the emitter. At higher temperatures a greater share of the energy appears in the lower wavelengths. The *Stefan-Boltzmann law* evaluates the energy radiated by a black body,

$$q = \sigma A T^4 \tag{4-34}$$

where q = radiant energy, Btu per hr
A = radiation area, sq ft
T = radiation-surface temperature, °R
$\sigma = 0.173 \times 10^{-8}$ Btu per hr-sq ft-°R^4

Hence

$$q = 0.173A\left(\frac{T}{100}\right)^4 \tag{4-35}$$

Actual materials are not perfect radiators or absorbers; to compare them with black bodies, we use the *emissivity* ratio,

$$\epsilon = \frac{\text{total emission power of actual body}}{\text{total emission power of } \textit{black body}}$$

with both bodies at the same temperature. Table 4-3 lists some typical values for some relevant materials. Emissivities of actual materials vary with temperature and wave length. A *gray body* has an emissivity less than 1 and has constant intensity at all wavelengths and temperatures—this is a reasonable assumption for many bodies.

As mentioned before, radiant energy incident on a body may be absorbed, reflected, or transmitted,

$$\alpha + \rho + \tau = 1 \tag{4-36}$$

where α = *absorptivity* = fraction of incident radiation absorbed
ρ = *reflectivity* = fraction reflected
τ = *transmissivity* = fraction transmitted

Usually ρ is zero for gases, and τ is negligible for solids and liquids.

Kirchhoff's law states that the emissivity and absorptivity of a substance are equal for radiation with equal temperatures and wavelengths.

4-12. Radiation-heat-transfer Calculations. Equation (4-34) when applied to the net interchange between two actual bodies is

$$q = 0.173AF_eF_a\left[\left(\frac{T_1}{100}\right)^4 - \left(\frac{T_2}{100}\right)^4\right] \tag{4-37}$$

where F_e = emissivity factor allowing for difference from black-body radiation for two bodies involved
F_a = factor accounting for effect of geometry of position of two bodies

We determine F_a mathematically and try to get the geometrical setup nearest to actual radiating conditions. Hottel gives values for this factor

TABLE 4-3. EMISSIVITY OF SURFACES

Surface	Temperature, °F	Emissivity ratio, ϵ
Aluminum:		
Highly polished	440–1070	0.039–0.057
Polished	73	0.040
Rough plate	78	0.055
Brass:		
Highly polished	494–710	0.033–0.037
Dull	120–660	0.22
Oxidized	390–1110	0.61–0.59
Copper:		
Commercial polish	66	0.030
High polish	176	0.018
Cast iron:		
Polished	392	0.21
Rusted	67	0.69
Rough and oxidized	100–480	0.95
Steel:		
Polished	212	0.066
Oxidized and rough	100–700	0.94–0.97
Molten	2730–3000	0.42–0.53
Monel metal, oxidized at 1110 F	390–1110	0.41–0.46
Asbestos	100–700	0.93–0.95
Brick:		
Red and rough	70	0.93
Building	1832	0.45
Magnesite refractory	1832	0.38
Plaster:		
Rough lime	50–190	0.91
Aluminum paint:		
Aluminum with varnish binder on rough plate	70	0.39
Other aluminum paints (variable)	212	0.27–0.67

for a variety of geometrical shapes of the surfaces involved in radiant-heat transfer.

Power-plant applications can in some cases be approximated by the following spatial relations for radiation between bodies 1 and 2. Data for Eq. (4-37) are shown.

Bodies 1 *and* 2 *are infinite parallel planes:*

$$A = A_1 = A_2 \qquad F_a = 1 \qquad F_e = \frac{1}{1/\epsilon_1 + 1/\epsilon_2 - 1} \tag{4-38}$$

Body 1 *is small and completely enclosed by large body* 2:

$$A = A_1 \qquad F_a = 1 \qquad F_e = \epsilon_1 \tag{4-39}$$

Body 1 *is large compared with completely enclosing body* 2:

$$A = A_1 \qquad F_a = 1 \qquad F_e = \frac{1}{1/\epsilon_1 + 1/\epsilon_2 - 1} \tag{4-40}$$

Body 1 *is a sphere or infinite cylinder enclosed by similarly shaped body* 2:

$$A = A_1 \qquad F_a = 1 \qquad F_e = \frac{1}{1/\epsilon_1 + (A_1/A_2)(1/\epsilon_2 - 1)} \tag{4-41}$$

For cases other than these, references must be consulted.

When heat leaves a surface both by radiation and convection, a combined coefficient is sometimes used.

$$h = h_r + h_c \tag{4-42}$$

where h = combined effect, Btu per sq ft-hr-°F
h_r = radiation effect, Btu per sq ft-hr-°F
h_c = convection effect, Btu per sq ft-hr-°F

Values for h_c can be obtained from convection-film equations (4-16) to (4-32). By solving Eq. (4-37) for q/A

$$h_r = \frac{q}{A(T_1 - T_2)} \tag{4-43}$$

and

$$R_r = \frac{1}{h_r}$$

where R_r is the thermal resistance for radiation.

4-13. Heat Exchangers. In power-plant heat exchangers heat from one fluid transfers to another. In giving up heat one fluid may cool or condense from vapor to liquid; in absorbing heat the other fluid may warm or evaporate from liquid vapor. Tubular or flat metal walls usually separate the fluids. Specific examples in power plants include steam generators, condensers, feedwater heaters, oil coolers, evaporators, hydrogen coolers, air heaters, superheaters, and economizers.

Heat-transfer calculations take account of the thermal resistance of fluid and films on each side of the wall and of the wall interior. When both fluids are gases, the wall resistance may be neglected. The over-all resistance depends on the flow pattern, fluids, and the presence of scale and fouling (Fig. 4-10). Equations (4-13) and (4-14) are transposed to

$$q = UA(t_s - t_r) \tag{4-13}$$

$$U = \frac{1}{\Sigma R} = \frac{1}{1/h_1 + 1/\alpha + 1/h_2} \tag{4-14}$$

In this equation A is arbitrary but all resistance must be based on the same area, either inner or outer; normally we use outside tube-surface areas. In using outer areas for a circular tube

$$\Sigma R = \frac{1}{h_1}\frac{D_2}{D_1} + \frac{r_2 \ln (r_2/r_1)}{k} + \frac{1}{h_2} \tag{4-44}$$

In using the inner area of circular tubes

$$R = \frac{1}{h_1} + \frac{r_1 \ln (r_2/r_1)}{k} + \frac{1}{h_2}\frac{D_1}{D_2} \tag{4-45}$$

where subscripts 1 and 2 refer to inner and outer annulus conditions. In Eq. (4-13) U is assumed constant over the heat-flow path. The fluid properties are assumed to correspond to the arithmetic average temperature of the range through which the fluid changes. Table 4-4 lists some typical film coefficients used in practice.

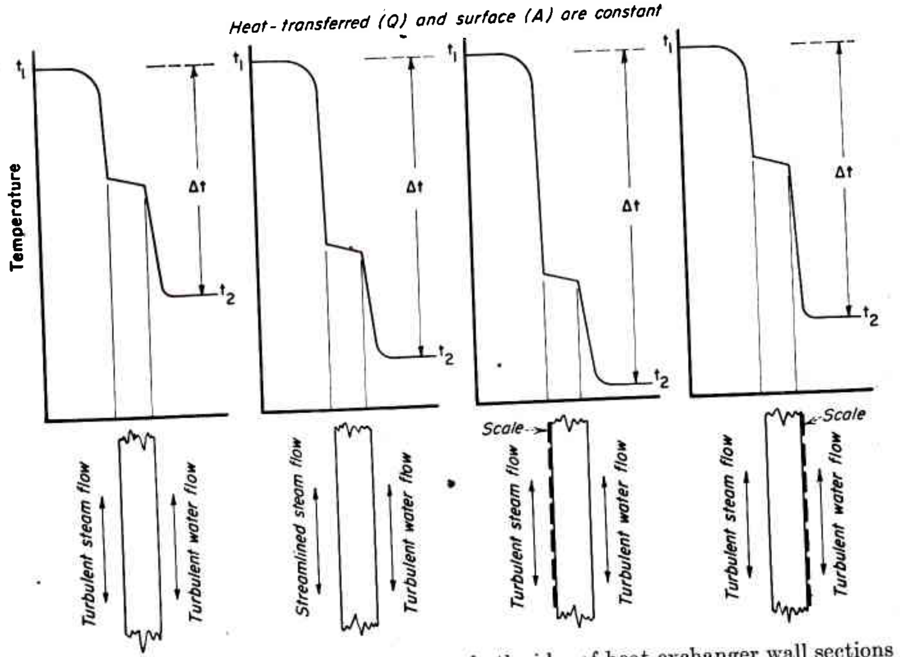


FIG. 4-10. Fluid-film resistances existing on both sides of heat-exchanger wall sections markedly affect temperature drops and temperature of wall surfaces; so does presence of scale or other deposits.

Since t_s and t_r may vary in heat exchangers, we have a problem in finding the value of $t_s - t_r$ to be used in Eq. (4-13). Figure 4-11 shows possible temperature variations in heat exchangers; note the significance of *parallel flow* and *counterflow*. The proper value of the mean temperature difference can be shown to be the *logarithmic mean temperature difference* (LMTD) computed as

$$(t_s - t_r)_{\text{mean}} = \text{LMTD} = \frac{\Delta t_{\max} - \Delta t_{\min}}{\ln (\Delta t_{\max}/\Delta t_{\min})} \tag{4-46}$$

TABLE 4-4. POWER-EQUIPMENT FILM COEFFICIENTS[a]

Heat-transfer process	*h, Btu per sq ft-hr-°F*
Condensing vapor:	
Steam	1000–2000
Ammonia	500–800
Light oils (volatile fuels)	200–300
Heavy oils	20–50
Evaporating liquids:	
Water	800–2000
Ammonia	400–500
Light oil	150–300
Heavy oil	10–50
Heating or cooling:	
Water	300–1500
Gases	5–250
Oils	10–120

[a] Courtesy of Foster Wheeler Corporation.

The LMTD applies to the four arrangements of Fig. 4-11. For exchangers having two constant temperatures as during boiling and condensing the mean temperature difference is their arithmetic difference. The two basic flow arrangements compare as follows:

Parallel flow:

1. Final temperatures of both fluids approach the same limit.
2. Effective for temperature control of wall material.
3. Transfers less heat than counterflow for same area and conditions.

Counterflow:

1. Permits greatest possible temperature change in each fluid.
2. Transfers more heat than parallel flow for same area and conditions.
3. Temperature of cooler fluid more nearly approaches the initial temperature of the hotter fluid.

4-14. Shell-and-tube Heat Exchangers. In this type of exchanger one fluid flows through parallel tubes arranged in a bundle between tube sheets and the other fluid over the tube exteriors but confined in a shell holding the bundle (Fig. 4-12). These designs use neither pure counterflow nor parallel flow but incorporate a counterflow arrangement in which the shell fluid passes over the tubes at right angles to their axes.

The average temperature difference may be computed by

$$(t_s - t_r)_{\text{mean}} = F\,\Delta t_{LMTD} \tag{4-47}$$

where F is the correction factor for the given shell-and-tube arrangement. Figure 4-13 shows factors for two types of exchangers. The Standards of the Tubular Exchanger Manufacturers Association (TEMA) contains graphs for many heater layouts.

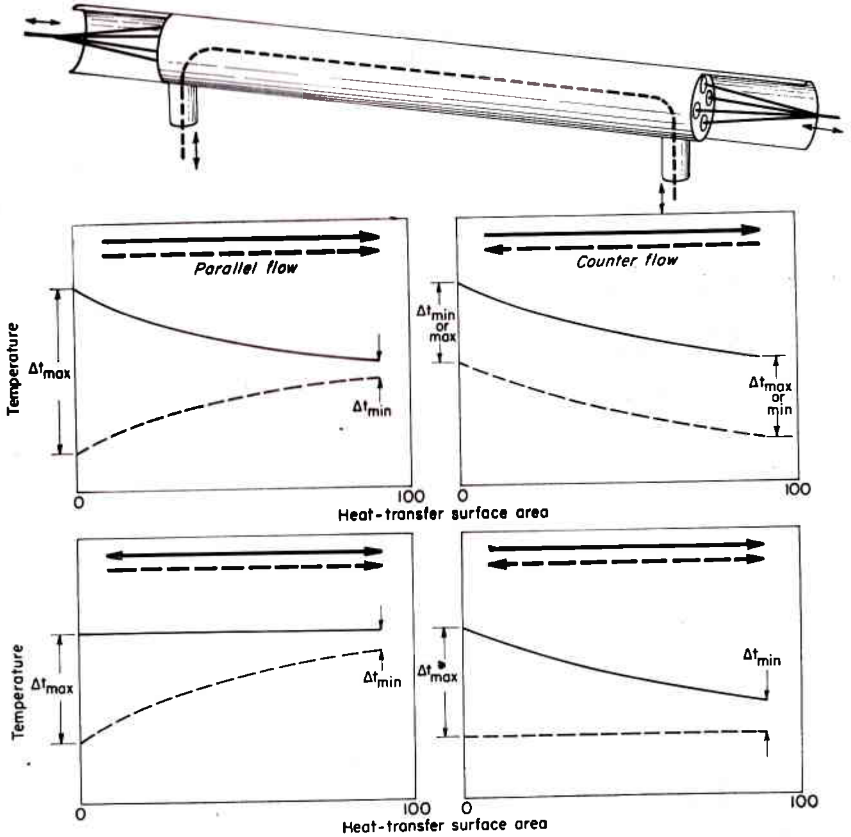


FIG. 4-11. Possible temperature patterns in heat exchangers. In upper graphs both hot- and cold-fluid temperatures change in passing through heater. In lower left hot fluid may be a condensing vapor; in lower right cold fluid may be a boiling saturated liquid. Splitting up fluid streams into several paths (top sketch) increases heat-transfer surface between them for given volume of exchanger.

4-15. Extended Heater Surface. Welding fins to tubes and plates substantially raises the total heat transferred across surfaces. They prove especially useful in transmitting heat from gases into a heat-exchange surface. This offsets the high film resistance by providing more surface in a given volume. Fins vary in width, cross section, and material; they may be integrally cast with, or attached to, the main surface by crimping or other methods.

4-16. Furnace Heat Transfer. In conventional power plants the combustion process releases the thermal power in a furnace. The complexity of the mass and heat-transfer effects makes practical furnace design highly empirical. This has produced highly efficient furnaces, but further improvements depend on a more basic knowledge of combustion and heat transfer in flames and fuels.

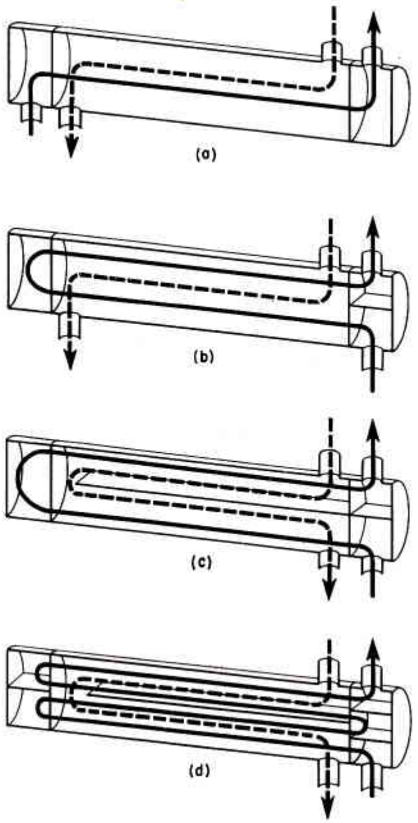


FIG. 4-12. Paths of fluids through shell and tube exchangers can be varied in many ways depending on the job to be done. (*a*) Single-pass tube and single-pass shell; (*b*) two-pass tube and single-pass shell; (*c*) two-pass tube and two-pass shell; (*d*) four-pass tube and two-pass shell. This last arrangement can be duplicated by putting two heaters like (*b*) in series for both tube-fluid and shell-fluid flows.

A modern furnace (1) burns fuel effectively, (2) transfers heat to water and steam primarily by radiation, and (3) allows water and steam to circulate. Fuel and air enter a furnace at ambient temperatures. Furnace action involves energy release by chemical reaction, heat transfer by radiation, conduction, and convection, and formation of combustion products. Only part of the chemically released energy transfers to the water and steam, and so the gases leave the furnace with a temperature of about 2000 F. They give up heat successively to heat-transfer surfaces of the superheater, economizer, and air preheater to enter the stack at about 250 to 600 F.

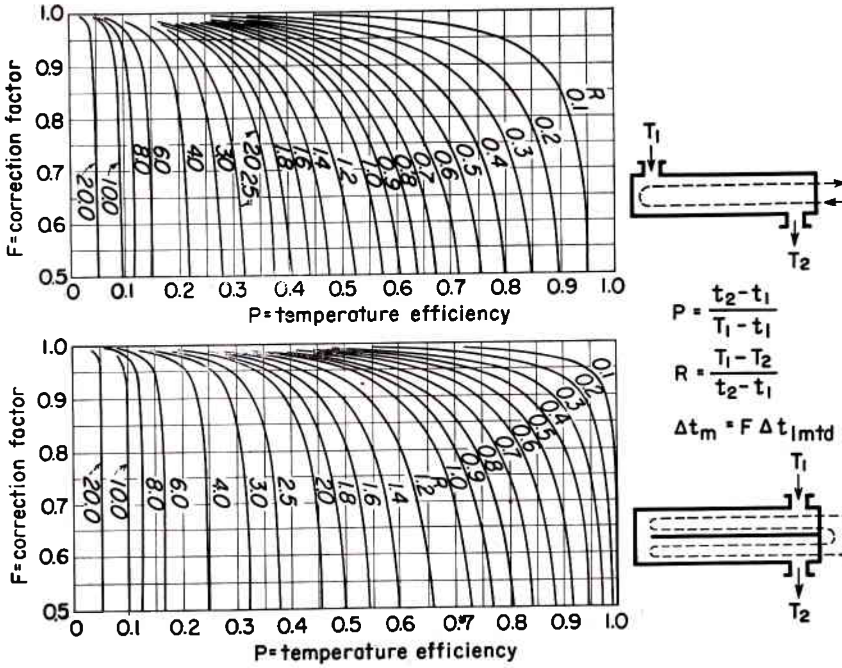


FIG. 4-13. When heater flow paths differ from true counterflow, a correction factor must be applied to the logarithmic mean temperature difference.

The over-all process can be summarized by a first-law equation,

$$E_c + H_{F+A} = Q_T + H_P \tag{4-48}$$

where E_c = chemical energy of fuel
H_{F+A} = enthalpy of mixture (fuel + air)
Q_T = heat transfer
H_P = enthalpy of products

Fuel heating values measure the energy released by the chemical reactions in the combustion process. Heating values depend on the fuel type, combustion process (constant-volume or constant-pressure), process temperatures, and the condensation of water vapor in the combustion gases. Higher values assume that all the water vapor condenses to release its heat of vaporization; lower values assume that the water vapor retains its state. Actual heat yields usually lie between these values.

Many studies have related furnace heat transfer to the radiant surface area, fuel combustion rate, and air-fuel ratio. These studies develop values for

$$\eta_F = \frac{Q_T}{Q_R} \tag{4-49}$$

where η_F = furnace absorption efficiency
Q_T = heat transferred per lb fuel, Btu
Q_R = available heat per lb fuel, Btu, evaluated as $E_c + H_{F+A}$

One of the first relations proposed by Hudson was later in its improved version known as the Hudson-Orrok equation. One form is

$$\eta_F = \frac{1}{1 + C(W_{fg}/Q_R)(Q_R/A)^{1/3}} \tag{4-50}$$

where C = constant depending on furnace contours, fuel, and walls
W_{fg} = flue-gas flow, lb per hr
Q_R = heat released in furnace, 1000 Btu per hr
A = projected area of radiant heating surface, sq ft

Other correlations have been made by Wohlenberg, Hurvick, Broido, and Konokov. These recognize the many variables of furnace heat transfer:

1. Fraction cold
2. Heat-release rate
3. Fuel—type, firing method, fineness of suspended particulate matter, heating value, flame shape and length, temperature, emissivity
4. Air-fuel ratio
5. Inlet-air temperature
6. Furnace-wall type
7. Furnace-wall condition—slag and ash characteristics
8. Geometry of furnace

Fraction cold depends on the water cooling in the furnace walls and measured as the ratio of water cooled to refractory surface; it may be evaluated by other ratios.

4-17. Friction Losses in Pipes and Ducts. As a gas or liquid flows through a pipe or duct, friction forces resist the flow and cause a pressure loss. The friction forces act at the wall and in the fluid interior. We must know these losses to design pumps and fans for power plants. For *incompressible* (liquid) flow the friction losses are

$$h_f = f\frac{LV^2}{4R_h 2g} \tag{4-51}$$

where h_f = head loss, ft of fluid flowing, caused by friction
L = length of duct or pipe, ft
V = mean flow velocity, ft per sec
R_h = hydraulic radius = (cross-section area)/(wetted perimeter)
f = dimensionless friction factor

In all cases

$$\Delta p_f = h_f \rho \tag{4-52}$$

where p_f = pressure loss, psf
ρ = fluid weight density, lb per cu ft

For circular pipes

$$4R_h = \frac{4\pi D^2}{4\pi D} = D \tag{4-53}$$

where D is the pipe diameter in feet. For laminar flow the friction factor is a function of the Reynolds number alone. For turbulent flow the friction factor depends on both Reynolds number and the geometry of the conduit surface. A parameter known as the *relative roughness* (ϵ/d) evaluates the geometry. Here ϵ indicates the surface roughness in pipe of diameter D. Figures 4-14 and 4-15 correlate the friction factor with the Reynolds number and surface roughness for new straight pipes. In other than circular-cross-section pipe use the hydraulic diameter in place of D to obtain N_{Re} and pressure drops,

$$D_h = 4\frac{A}{P}. \tag{4-54}$$

where D_h = hydraulic diameter, ft
A = cross-section area, sq ft
P = wetted perimeter, ft

Figure 4-16 helps calculate pressure losses in elbows, bends, and valves. These are given in terms of equivalent lengths of straight pipe run. The equations above give only friction-head losses in horizontal pipe. Corrections account for inclined or vertical pipe runs that change the potential head. Change in velocity head must be allowed for.

The effects of flow on the surface finish of pipes and ducts cannot be accurately predicted. Scale, corrosion, or other deposits may radically alter the relative roughness. New designs usually have safety margins of extra capacity. Existing systems can be measured directly to find pressure loss.

The above equations are often applied for *compressible flow* (gases and vapors), when pressure drops and fluid temperatures do not cause large density changes. Limits lie in density changes of less than 7 per cent or pressure drops of less than 10 per cent of initial pressure with constant fluid temperature. Another method divides the pipe into sections and

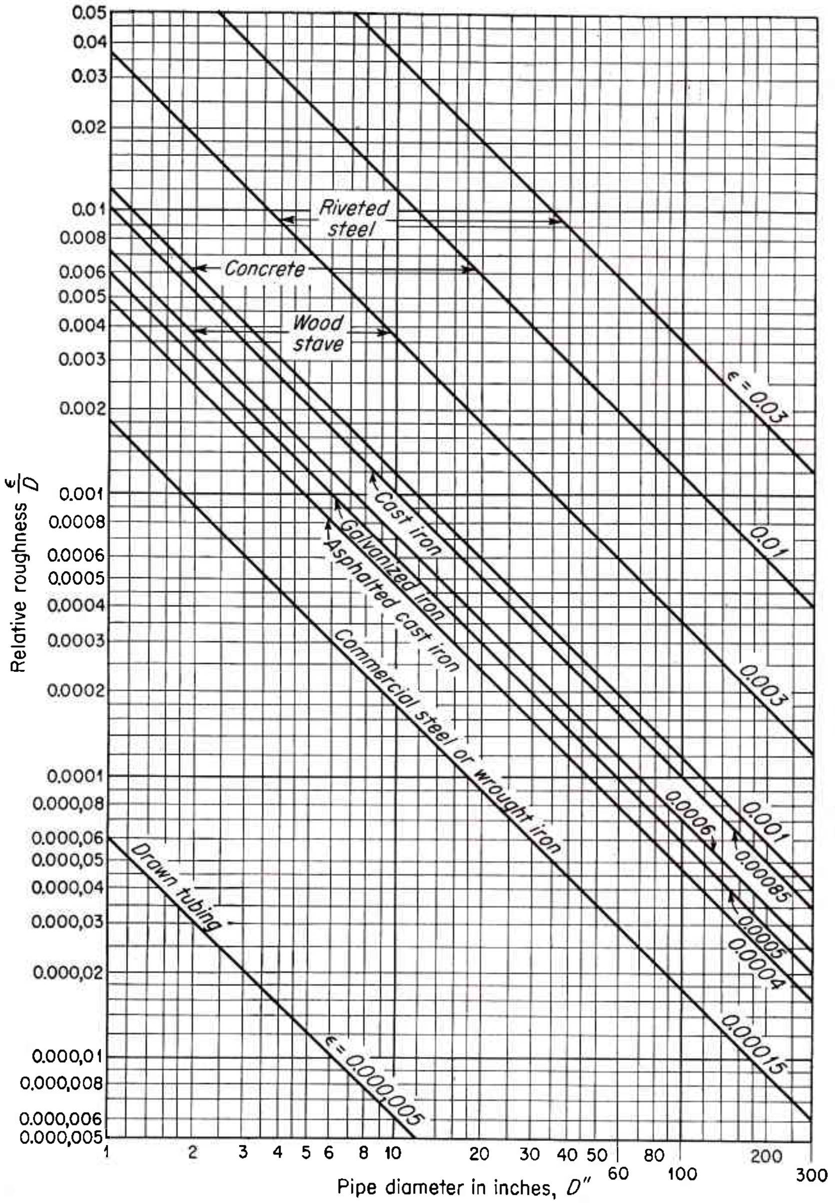


FIG. 4-14. The type of pipe and its diameter are used in this chart to obtain a "relative roughness" ϵ/D. (*Courtesy of L. F. Moody, Friction Factor for Pipe Flow, Transactions of the ASME,* 1944.)

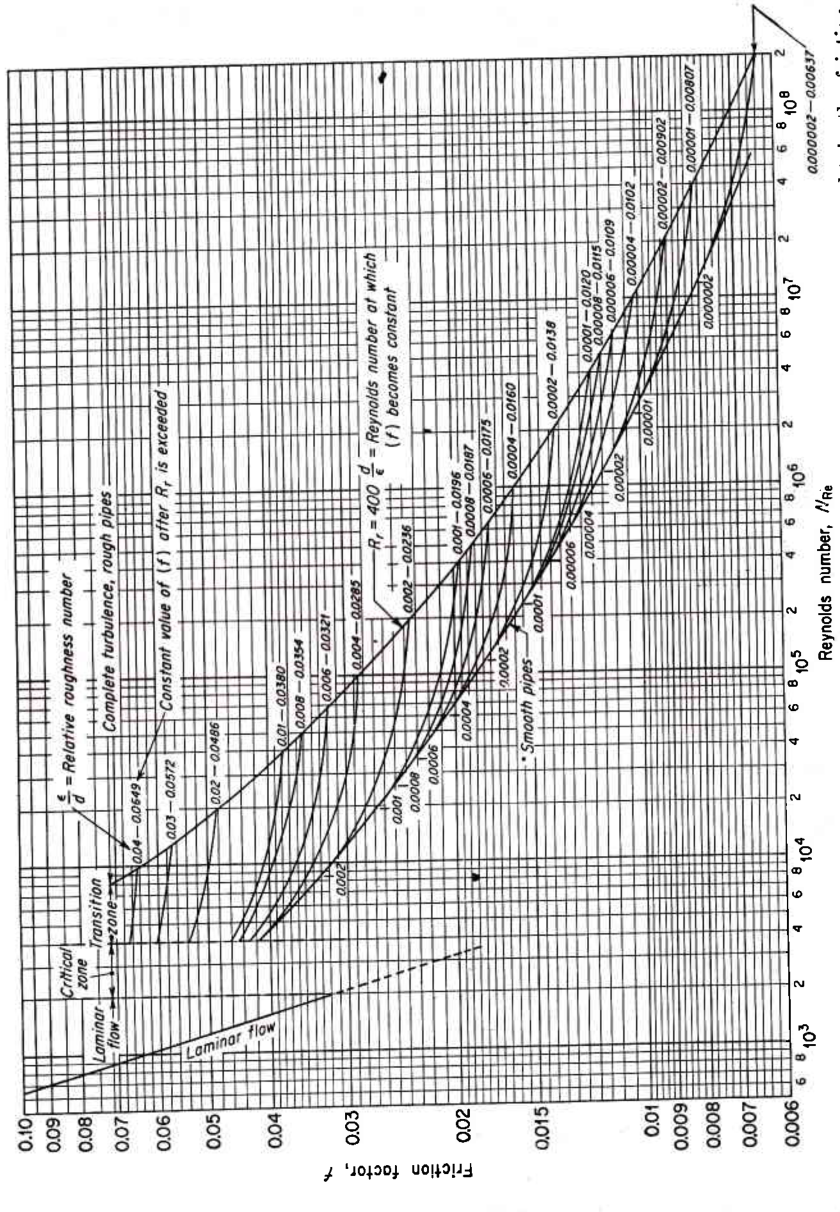


FIG. 4-15. The Reynolds number of flow and the relative roughness are used in this chart to obtain the friction factor f. Note that ϵ is the measure of absolute roughness of conduit. (*From ASME Paper No.* 56-A-63, *Losses in Pipe and Fittings, by R. J S. Pigott.*)

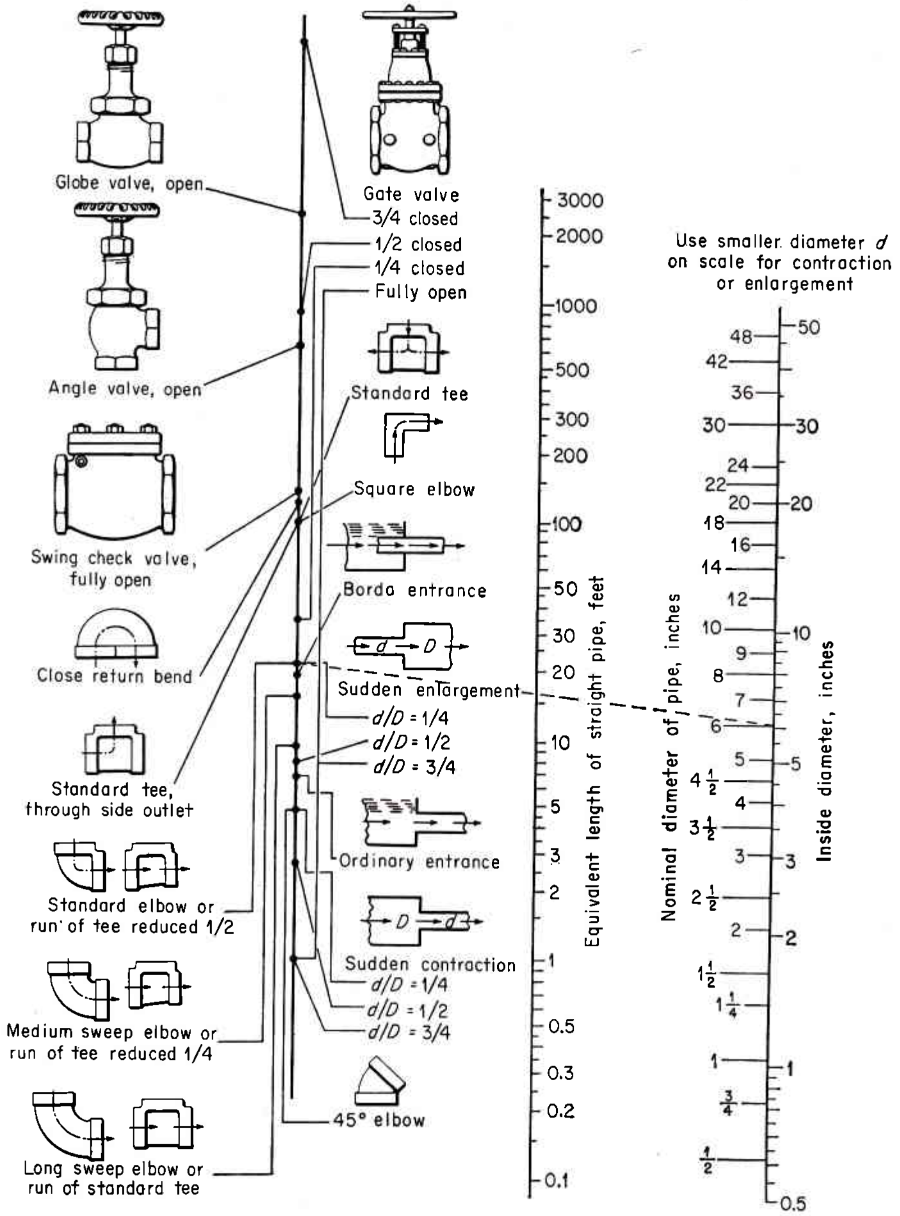


FIG. 4-16. This chart shows the equivalent resistances of various fittings and forms in pipes and ducts, expressed as equivalent feet of straight-run pipe or ducts. (*Courtesy of Crane Company.*)

figures pressure drop for each section. Compressibility must be considered in gases and vapor when they flow in long pipes.

PROBLEMS

4-1. Compare the steady-state heat-conduction characteristics of (*a*) magnesite refractory brick; (*b*) ½Cr-0.5Mo steel; (*c*) 85 per cent magnesia insulation. Calculate the heat loss (1) across each material, in Btu per square foot per hour, for parallel flow; (2) across a series flow (refractory-magnesia-steel). Outside-wall temperatures are 1000 and 60 F, respectively for each case, and a 3-in. thickness is used for each.

4-2. A flat wall consists of 12 in. thickness of a 90 per cent alumina refractory brick and 8 in. of 85 per cent magnesia. Hot- and cold-face temperatures are 1870 and 200 F. (*a*) Plot a temperature-resistance diagram. (*b*) Plot a curve of interface temperatures as the magnesia thickness is varied from 8 to 0 in.

4-3. A furnace wall consists of a 12-in. lining of magnesite refractory brick and a 5-in. thickness of 85 per cent magnesia with a steel casing 0.10 in. thick. Inside-furnace gas temperature is 2200 F, and the surroundings are at 80 F. Gas-side film coefficient is $h_1 = 15$ Btu per sq ft-hr-°F and air-side $h_2 = 4.0$. Determine: (*a*) the thermal current q/A; (*b*) interface temperatures; (*c*) effect on thermal current and inside refractory-wall temperature if the magnesia insulation were doubled.

4-4. A furnace wall consists of 12 in. lining of chrome refractory brick and 6 in. of 2900 F insulating firebrick. Surface temperatures are 2800 and 250 F. Assuming no change in surface temperatures, what effect will the addition of 4 in. of diatomaceous insulation have on (*a*) thermal current; (*b*) interface temperatures?

4-5. A steel tube (1.0Cr-0.5Mo) with 1.315 in. OD and 1.049 in. ID transmits 38,000 Btu per hr-sq ft of outside wall to the surroundings. Calculate the outside-wall temperature if the inner wall is at 212 F.

4-6. A 5-in. pipeline (nominal) carries saturated steam at 400 F. It has a covering of 3 in. of 85 per cent magnesia and 2 in. of cork. Flow conditions give film coefficients $h_1 = 2500$ Btu per hr-sq ft-°F and $h_2 = 2.0$ with an outside-air temperature of 70 F. Draw the temperature-resistance diagram, and from it determine the interface temperatures.

4-7. A 2½-in. pipeline (nominal) carries saturated steam at 200 psig and is covered with 2 in. of a diatomaceous-type pipe covering. Calculate: (*a*) heat loss per 100 ft of pipe and h_2 if the lagging surface temperature is 110 F (assume inside film and pipe metal resistance negligible); (*b*) outside film coefficient h_2 for a room temperature of 65 F; (*c*) annual steam cost saving per 100 ft of pipe due to using insulation (assume that steam costs *C* dollars per 1,000 lb and that h_2 for bare pipe = 6.0 Btu per hr-sq ft-°F).

4-8. An 8-in. steel pipeline (nominal) carries steam at 450 F. An 85 per cent magnesia pipe covering is to be applied of such a thickness so as to limit the surface temperature to 120 F with room temperature 60 F. Assume inside and outside coefficients 3000 and 2.0 Btu per hr-sq ft-°F and k (for steel) = 24 Btu per hr-sq ft-°F-ft. Calculate the magnesia thickness in inches.

4-9. A boiler tube for steam at 1,200 psia is 3½ in. OD and 2.71 in. ID. Assume an internal film coefficient of 2000 Btu per hr-sq ft-°F and a thermal flow of 50,000 Btu per hr-sq ft based on outside area. Calculate: (*a*) outside tube temperature; (*b*) allowable boiler scale thickness if the metal is not to exceed 900 F. Assume that k (for steel) = 25 Btu per hr-sq ft-°F-ft and k (for scale) = 0.3.

4-10. An oil cooler is made of 0.75-in. copper tubes of No. 18 Bwg. Cooling water

inside tubes is 60 F and oil temperature 220 F. Water-side convection film coefficient $h_1 = 193$ Btu per hr-sq ft-°F, oil side $h_2 = 75$. Compare the heat flow in Btu per hour per square foot of the tubes (*a*) clean; (*b*) water side scaled with 0.02-in. hard scale ($k = 0.35$ Btu per hr-sq ft-°F-ft).

4-11. A 1½-in. iron pipe carries water at 250 F in a high-temperature water system and discharges through an enlarging coupling to a 6-in. line. What flow pattern exists for a flow of (*a*) 5,500 lb per hr-sq ft of smaller pipe and (*b*) 25,000 lb per hr-sq ft of smaller pipe?

4-12. Prove that the Grashof, Stanton, and Prandtl numbers are dimensionless.

4-13. Calculate the film coefficient of heat transfer for heating water flowing through a straight tube. Average film temperature is 70 F, velocity is 1.6 ft per sec, and tube is 1.0 in. OD, No. 16 Bwg, and is 25 ft long.

4-14. A condenser tube is 15 ft long with a ⅞-in. OD and a No. 16 Bwg wall thickness. Water temperature averages 65 F and flows at 12 ft per sec. Calculate the heat-transfer film coefficient.

4-15. Air flowing normal to a deep staggered bank of tubes is at atmospheric pressure and is being heated from 60 to 500 F. Maximum flow velocities of 15 ft per sec occur. Determine the film coefficient of heat transfer if 3½-in.-OD tubes are used.

4-16. A plate type of air heater is 25 by 5 ft in cross section and handles flue gas at an average temperature of 440 F. Plates are parallel to the 5-ft side and spaced 1 in. apart. Flue-gas velocity is 10 ft per sec with a density of 0.05 lb per cu ft. Approximately what film coefficient may be expected?

4-17. Compare the convection heat-transfer coefficient for air and for steam flowing inside 5-in. (nominal) pipes both at an average temperature of 440 F and 25 psia with a velocity of 3 ft per sec.

4-18. Steam at 1 in. Hg abs is condensing on condenser tubes of ⅞ in. OD and No. 14 Bwg thickness. What is the temperature drop across the condensing film if $h = 1600$ Btu per sq ft-hr-°F and the water averages 60 F temperature?

4-19. Steam in a heater is condensing on steel tubes 25 ft long with 1⅛ in. OD and 0.12 in. wall thickness. Water temperature averages 320 F, and a mass current of 1,200,000 psf per hr occurs with a Reynolds number of 109,000. (*a*) Determine the various heat-transfer resistances referred to the inner tube wall. (*b*) Discuss the controlling resistance.

4-20. An uninsulated 4-in. (nominal) rough steel pipe is run through a square tunnel of red brick 8 in. on a side. Saturated steam is in the pipe, and a surface temperature of 320 F is expected with the brick temperature at 70 F. Calculate the radiant heat loss per 100 ft of pipe.

4-21. A 3-in. (nominal) pipe is covered with 1 in. of asbestos and runs through a large room with concrete-block walls at a temperature of 90 F. Plot the relation between radiant heat loss from the pipe for surface temperatures ranging from 200 to 600 F.

4-22. A 5-in. (nominal) steel pipe carrying saturated steam at 200 psia is covered with asbestos and aluminum paint with an emissivity value of 0.55. Surface temperature is 150 F, and the pipe runs vertically through a room 30 ft high with 70 F room temperature. Determine: (*a*) the radiant heat loss; (*b*) the convection heat loss; (*c*) the combined radiation-convection film coefficient.

4-23. An outside wall of a steam generator 15 by 30 ft is located near the building wall. The room wall is at 85 F and the steam-generator wall at 110 F. Emissivities are 0.85 and 0.80, respectively. (*a*) Compute the radiant heat loss. (*b*) If the loss is to be reduced 50 per cent by treatment of the building wall, what emissivity must the surfacing produce?

4-24. A horizontal, bright-tinned, thin iron duct carries hot air with 160 F duct temperature through a room at 80 F. An asbestos covering of 1 in. is proposed to reduce heat loss. Emissivities of the order of 0.06 and 0.94 are expected for the tinned duct and asbestos. (*a*) Is the proposal economically advantageous? (*b*) What effect would result from painting the duct?

4-25. Assume a $\frac{7}{8}$-in.-OD No. 18 Bwg Muntz-metal condenser tube ($k = 71$). Calculate the thermal resistance based on outside area (*a*) assuming the tube a thick wall; (*b*) assuming the tube a plane wall; (*c*) assuming the plane wall corrected for curvature by logarithmic mean diameter.

4-26. A blowdown heat exchanger from a boiler at 1,200 psia cools blowdown water to 600 F. Feedwater entering at 420 F and leaving at 500 F is used for heat recovery. Assume a counterflow exchanger, and calculate (*a*) LMTD; (*b*) arithmetic mean temperature difference.

4-27. A closed feedwater heater has the following design characteristics:

Feedwater, 350,000 lb per hr, 78 F in, 144 F out
Exhaust steam, h = 1120 Btu per lb, p = 5.5 psia
Construction, two-pass 1,920-sq ft $\frac{5}{8}$-in.-OD No. 18 Bwg tubes of Admiralty metal

Calculate: (*a*) extraction steam rate; (*b*) LMTD; (*c*) over-all U based on outside tube area with a water side h = 1700 Btu per hr-sq ft-°F.

4-28. A heat exchanger is designed to heat water from 120 to 320 F at a 60,000 lb per hr rate. Hot gases cooling from 900 to 550 F are used as the heat source. An over-all coefficient U = 15 Btu per hr-sq ft-°F is expected. Compare the areas needed for (*a*) counterflow; (*b*) parallel flow.

4-29. A four-pass closed feedwater heater handles 298,240 lb per hr of water heated from 312 to 375 F, using saturated steam at 370 psi and negligible subcooling of the condensate. Calculate the over-all heat-transfer coefficient if the tube area is 3,100 sq ft.

4-30. An oil cooler uses inlet water at 120 and 150 F outlet to cool 15,000 lb per hr of oil from 185 to 165 F. A two-pass design with oil in the shell is used, and an over-all U = 70 Btu per hr-sq ft-°F is expected. Calculate: (*a*) gallons per minute of cooling water; (*b*) heat-transfer area required.

4-31. Data from a condenser design are as follows:

Tubes, Admiralty metal, 25 ft long, single-pass design, $\frac{3}{4}$-in.-OD, No. 18 Bwg
Cooling water, 60 F inlet, V = 9.5 ft per sec
Heat-transfer data, h (condensing steam) = 1600, h (cooling water) = 1820 Btu per hr-sq ft-°F

Estimate the condenser pressure obtainable for a 10-F terminal temperature difference.

4-32. A tubular regenerator for a gas-turbine unit heats 420 lb per sec of air from 580 to 750 F. Turbine exhaust gas enters at 910 F and leaves at 720 F. Assume h (air side) = 12, h (gas side) = 10.5 Btu per hr- sq ft-°F. Calculate: (*a*) outside tube area (4-in.-nominal tubes); (*b*) maximum metal temperature of the tube metal for counterflow design.

4-33. Data for a two-pass surface condenser are as follows:

Cooling water, 1,200,000 lb per hr, 70 F entering, 82 F leaving, V = 5.0 ft per sec
Exhaust steam, p = 1.5 in. Hg pressure
Heat-transfer data, h (condensing steam) = 2100 Btu per hr-sq ft-°F, Muntz-metal tubes, $\frac{3}{4}$-in.-OD, No. 16 Bwg

Calculate the length of the tubes.

4-34. An air heater of the tubular type is designed with steel tubes of 3-in. (nominal) diameter with flue gas inside the tubes. Film coefficients for the gas and air side are 5.0 and 6.8 Btu per hr-sq ft-°F, respectively. Gas enters at 650 F and leaves at 365 F, while air enters at 60 F and leaves at 475 F. Counterflow conditions are approached in the design. Compute the outside tube area required per pound of air heated.

4-35. A single-pass condenser 20 ft long is designed with $\frac{7}{8}$-in.-OD No. 18 Bwg Admiralty-metal tubes. Exhaust steam is at 1.0 in. Hg, water velocity 7.0 ft per sec in the tubes, inlet water 56 F, and a steam-side coefficient of 1700 Btu per hr-sq ft-°F is expected. Calculate the number of tubes required for a terminal temperature difference of 10 F between entering steam and leaving cooling water with an exhaust flow of 150,000 lb per hr of steam at a quality of 90 per cent.

4-36. An economizer is designed to heat 300,000 lb per hr of feedwater from 425 to 510 F, using 260,000 lb per hr of combustion gases entering at 975 F and cooling to 650 F. The general design uses in-line horizontal tubes and a vertical gas flow with tubes 18 ft long, 30 tubes wide, 1.9 in. OD, 0.40 in. thick and a mass current of 2,850 lb per sq ft-hr across each tube section. Based on counterflow-heat-transfer calculations and any needed assumptions, determine the number of tubes high and the tube spacing for the unit.

4-37. A straight run of steam line is 200 ft long with a 3-in. (nominal) steel line. Steam at entrance is at 250 psia and dry. Estimate the pressure loss in pounds per square inch at a flow velocity of 5,000 ft per min.

4-38. A single-pass heat exchanger has 200 tubes of 1.25-in. No. 16 Bwg drawn tubes 15 ft long. Water enters at 200 F and is heated to 350 F at a flow velocity of 3 ft per sec. Compute the pressure loss in pounds per square inch.

4-39. A steel breeching 2 by 6 ft in cross section carries flue gas at a velocity of 20 ft per sec and an average density of 0.05 lb per cu ft and average temperature of 440 F. Estimate the pressure drop per 10 ft of straight-run breeching in inches of water.

4-40. Water at 60 F flows through a line 100 ft over-all of 2-in. (nominal) diameter steel pipe with a velocity of 10 ft per sec. The line has a globe valve full open, a gate valve half closed, and four medium sweep elbows in series. Estimate the pressure drop in pounds per square inch.

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PART II

STEAM POWER PLANTS

SECCION CENTRAL HIDRAULICA
U.T.E.
RINCON DEL BONETE

CHAPTER 5

FUEL-BED FIRING

5-1. Practical Combustion. Fuels burn in a confined space called a *furnace*. Rate of supply of both fuel and air to the furnace must be controlled to release efficiently the heat developed by combustion (see Chap. 2). The furnace must direct the energy released by the fire to the heat-transfer surfaces of the steam generator: waterwalls, convection tubes, superheater, reheater, economizer, and air preheater.

An efficient furnace receives the air and fuel prepared for burning and ignites them, and the fire completely consumes all the combustible elements in the fuel. The walls of the furnace must be able to withstand the high temperatures of the fire. Walls may be made of refractory materials or water-filled steel tubes or a combination of both. High-temperature gases from the fire leave the furnace to enter the gas passes of the boiler.

The method of feeding fuel and air to a furnace depends on the physical form of the fuel. *Solid fuels* such as coal, coke, wood, and refuse need a grate at the furnace bottom to hold a bed of fuel. Some form of stoker mechanism places the fuel on the grate automatically. Air enters under the bottom of the grate and passes through holes in it and the fuel bed. The air reacts with the hot fuel bed as it passes through and releases the stored chemical energy by combustion.

Pulverized fuels enter a furnace through openings called *burners*. These burners receive the powdered fuel carried by a rapidly moving stream of *primary air*. Additional *secondary air* joins the fuel flow in the burner, and the mixture jets into the furnace. The flame in the furnace ignites the fuel, and it burns in suspension.

In using *oil* as a fuel the burner breaks up or atomizes the oil into minute droplets and mixes it with air. The mixture blows into the furnace, ignites, and burns. A burner handles a *gas fuel* in much the same manner, except that gas needs no atomization. The term *burner* is a misnomer: these devices act as mixers, and burning takes place in the furnace.

5-2. Combustion Conditions. Four conditions must be established to burn fuels properly: (1) high flame temperature, (2) turbulent mixing of air and fuel, (3) the correct ratio of fuel and air, and (4) enough time to complete the burning process.

1. Furnace flame temperature must be high enough to ignite the incoming raw fuel and air. Ignition raises the temperature of both fuel and air to the level where they quickly react with each other to produce heat. The process needs a finite time. Sometimes supplying air at too great a rate can blow out a fire by cooling the fuel and stopping ignition. Table 5-1 lists the ignition temperature of various fuel elements. The energy of ignition comes from fuel already burning in the furnace.

TABLE 5-1. COMBUSTIBLE IGNITION TEMPERATURES

Combustible element	*Ignition temperature, °F*
Fixed carbon in bituminous coal	770
Fixed carbon in semibituminous coal	870
Fixed carbon in anthracite coal	925
Acetylene (C_2H_2)	900
Ethane (C_2H_6)	1000
Ethylene (C_2H_4)	1020
Hydrogen (H_2)	1130
Methane (CH_4)	1200
Carbon monoxide (CO)	1210
Sulfur (S_2)	470

2. Air and fuel must be thoroughly mixed to burn the fuel completely. Various arrangements of burners do this with pulverized coal. In fuel-bed burning part of the combustion air may be injected into the furnace at high speed across the direction of normal flow of the combustion gas. This turbulently mixes the air and furnace gases being burned. Without this turbulence furnace gases may stratify: streams of combustible gases and air may flow side by side without reacting with each other.

3. Rate of supply of both air and fuel must be closely regulated to achieve the proper fuel-air ratio or excess air. Automatic combustion control systems do this in modern steam generators. The optimum fuel-air ratio may be that value at which no combustible gases appear in gases leaving the furnace. This usually coincides with the highest furnace temperature. Sometimes furnace temperatures must be lowered by additional excess air to reduce maintenance of the furnace and its associated equipment. The dry-gas loss usually drops to a minimum when the furnace temperature reaches a maximum.

4. Sufficient time must be allowed to complete the burning process in the furnace. This is primarily a matter of properly proportioning the furnace dimensions. In general, the taller the furnace, the longer the time available for completing combustion before the gases reach the boiler convection surfaces.

5-3. Fuel-bed Combustion. Solid coal may burn on a grate by two methods: (1) *overfeed* or (2) *underfeed*.

Let us first study the overfeed mechanics of combustion (Fig. 5-1). The air with its water-vapor content (humidity) comes from the atmosphere. A forced-draft (f-d) fan pressurizes the air slightly before it enters under the bottom of the grate. The air consists of N_2, O_2, and H_2O. In flowing through the grate openings the air is heated while it cools the grate. The warmed air then passes through a layer of hot ashes and picks up additional energy. The region immediately above the ashes contains a mixture of incandescent coke and ash, coke content increasing in the upward direction. As the air contacts the incandescent coke, the O_2 reacts with the C to form CO_2. In this part of the fuel bed the rate of carbon oxidation depends entirely on the rate of air supply. Increasing the air flow raises the rate of energy release by increased burning. For a normal fuel bed more than 3 in. deep all the free O_2 in the air disappears in the incandescent region. Water vapor entering with the air reacts with the incandescent coke to form CO_2, CO, and free H_2. Upon further travel through the incandescent region some of the CO_2 reacts with coke to form CO. Gases leaving the incandescent region consist of N_2, CO_2, CO, H_2, and H_2O; no free O_2 will be present (unless there are large holes in the fuel bed).

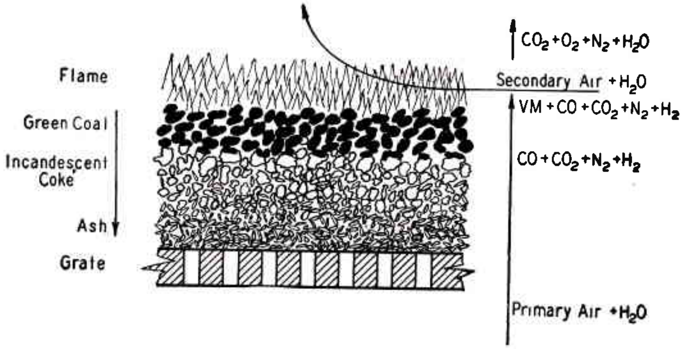


FIG. 5-1. Overfeed fuel-bed section receives fresh coal on top surface; ignition plane lies between green coal and incandescent coke.

Fresh fuel undergoing distillation of its volatile matter forms the topmost layer of the fuel bed. Heat for distillation and eventual ignition comes from four sources: (1) by conduction from the incandescent coke below, (2) from high-temperature gases diffusing through the surface of the bed, (3) by radiation from flames and hot gases in the furnace above, and (4) from the hot furnace walls. The ignition zone lies directly below the raw fuel undergoing distillation.

Gases leaving the surface of the fuel bed contain volatile matter of the raw fuel and N_2, CO_2, CO, H_2, and H_2O. To burn the combustible gases, additional secondary air must be fed into the furnace to supply the needed oxygen. The secondary air must be injected at considerable speed to create turbulence and to penetrate to all parts of the area above the fuel bed. The combustible gases then complete burning in the furnace. Combustion gases entering the boiler passes then contain N_2, CO_2, O_2, and H_2O, plus some CO if burning is incomplete.

Fuel, coke, and ash in the fuel bed "move" in a direction opposite to that of the air and gases. Raw fuel continually drops on the surface of the bed. Here it loses its volatile matter by distillation, while its fixed carbon and ash heat to incandescence. During incandescence the fuel continuously loses its carbon by oxidation until only the ash remains. The rising air feed cools the ash until it finally rests in a plane immediately adjacent to the grate.

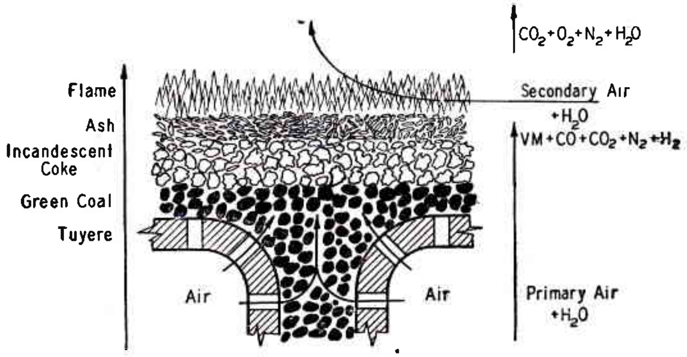


FIG. 5-2. Underfeed fuel-bed section receives fresh coal from retort underneath between two tuyeres; ignition plane lies between green coal and incandescent coke.

Now let us study the underfeed method of burning fuel on a grate (Fig. 5-2). Air, after passing through holes in the tuyere (grate), meets raw coal (green coal). Diffusing through spaces in the raw fuel, it then meets fuel losing its volatile matter by distillation. Heat for distillation comes by conduction through the mass of the fuel bed from the region of incandescent coke above. The air mixes with the distilled volatile matter, passes through the ignition zone, and enters the region of incandescent coke. Heat for ignition also comes from the incandescent coke above, counter to the direction of flow of air and gases.

The same reactions take place in the incandescent zone as in the overfeed method, but with some breaking up of the molecular structure of the volatile matter and partial reaction with O_2 of the air. The gases then pass through a region of incandescent ash on the surface of the fuel and finally discharge to the furnace with substantially the same constituents as for overfeed. There is the same need for a supply of secondary air to complete burning the gases in the furnace.

Underfeed firing works best for semibituminous and bituminous coals

high in volatile matter and with caking tendency. The volatile matter passes through the incandescent region to heat to furnace temperature. So when mixed with secondary air it burns quickly. In overfeed burning the volatile matter will be somewhat cooler than the furnace gases and so need a slightly longer time to ignite and complete burning. This may create a tendency to form smoke.

Rate of air feed in underfeed burning has a marked effect on speed of ignition. The rising air will have a cooling effect on the raw fuel prior to ignition. At low rates of air flow heat conduction downward through the fuel bed proceeds faster than the removal of heat by the air. This lets the ignition plane move down a maximum below the bed surface, and creates a maximum depth of incandescent zone. As air-flow rate rises, however, it removes heat from the fuel (being warmed by conduction) at a more rapid rate and raises the ignition zone nearer the fuel-bed surface, narrowing the zone of incandescent fuel. Air flow can be increased to such a rate that it literally blows out the fire. This action cannot take place in the overfeed method since the air must first pass through the incandescent zone before reaching the ignition plane.

Air pressure under the grate controls the rate of air feed to the fire, the rate of burning rising almost directly with pressure increase. As air pressure rises, it exerts more and more force, tending to lift the fuel bed off the grate, making it necessary to carry a thicker fuel bed at higher rates of heat release. At higher air pressures than normally used there is a narrow range where the fuel bed tends to float and become fluid in character. In this condition the bed covers up the thin spots, tending to keep a uniform resistance to air flow through all sections. Increasing the air pressure slightly above this range or disturbing it by poking lifts the whole bed off the grate and ruins the fire.

5-4. Traveling-grate Stokers. These stokers have two forms: (1) chain grate and (2) bar grate. These differ only in the details of grate construction. Figure 5-3 shows the general arrangement of either type of traveling-grate stoker. The grate resembles an endless belt running over two sets of sprocket wheels. The upper level of the grate carries the fuel bed, moving it from front to rear of the furnace.

The grate surface of a *chain-grate stoker* is made up of a series of cast-iron links connected by bars or pins to form an endless chain, and a surface as wide as needed. When the links pass over the end sprockets, they move with a scissorlike action in respect to each other, which helps to dislodge any clinkers that may adhere to the grate.

In a *bar-grate stoker* (also called "traveling-grate stoker") the surface is made up of a series of cast-iron sections mounted on carrier bars. The carrier bars ride on two or more endless-type drive chains and may receive additional support from skid rails to carry the weight of the fuel bed.

In passing under the coal hopper a traveling grate takes on a layer of coal. A coal gate at the rear of the coal hopper regulates the depth of layer (fuel bed). The gate can be raised or lowered as needed. Primary air from the forced-draft fan discharge enters the wind box between the two layers of the grate. Secondary air enters the furnace through nozzles from ducts in the furnace walls not shown in the figure. Simultaneous adjustment of grate speed, fuel-bed thickness, and air flow controls the burning rate so that nothing but ashes remains on the grate by the time it reaches the furnace rear. The ashes fall into the ashpit as the grate turns on the rear sprocket to make the return trip on the lower level to the furnace front.

As the raw, or green, coal on the grate enters the furnace, the surface coal ignites from heat of the furnace flame and from heat reflected by the ignition arch. The ignition plane and incandescent zone move down through the incoming fuel bed. The ignition plane is inclined so that it intersects the grate surface a short distance into the furnace. The incandescent zone extends from the ignition plane to a similarly inclined plane beyond which nothing but ash remains on the grate. The fuel bed becomes thinner toward the furnace rear as the combustible matter burns off.

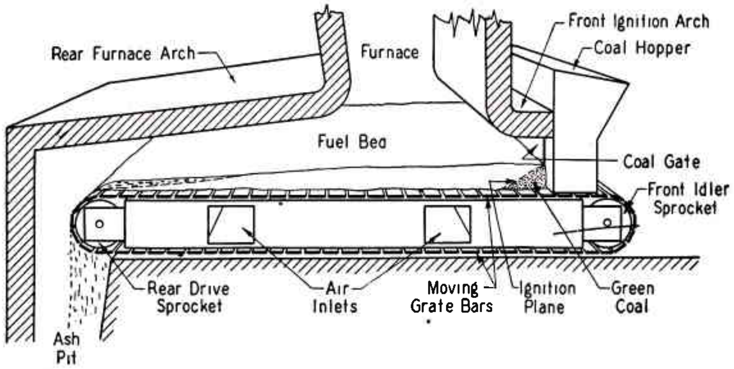


FIG. 5-3. Traveling-grate stoker moves like endless belt at slow rate, carrying fuel bed from right to left. Fuel-bed thickness is controlled by coal gate at back of coal hopper.

The fuel-bed thinning requires lowering of under-grate air pressure as the bed moves through the furnace. The wind box under the grate is divided into four sections fed by the two air inlets. Dampers in the sides of these inlets, under the grate, progressively lower the air pressure in the sections from front to stoker rear, even though all the air comes from just one fan at constant pressure.

Rear furnace arches covering the stokers help to: (1) maintain a high fuel-bed temperature, (2) reduce the fuel-bed fines leaving with the furnace gases, and (3) mix the combustible gases and air to complete burning. Secondary air injected at the neck of the furnace entrance aids in mixing the gases and supplies oxygen to complete combustion.

Traveling-grate stokers can burn any type of free-burning coal. Slightly caking coal can be burned with a high enough under-grate air pressure. Distinctly caking coals cannot be burned because such a fuel bed must be agitated.

Traveling-grate stokers can burn bituminous coal with 10 per cent moisture and 9 to 12 per cent ash at maximum rates of 500,000 Btu per sq ft-hr. When the moisture rises to 20 per cent and ash to 20 per cent, the maximum continuous burning rate drops to 425,000 Btu per sq ft-hr. Anthracite may be burned at a maximum continuous rate of about 400,000 Btu per sq ft-hr. But as the percentage of fines rises to more than 1 per cent through 200-mesh, the rate drops to 325,000 Btu per sq ft-hr.

Preheated air must be used with caution. Some bituminous that burns without caking at 80 F may mat over when air rises to 250 F. Some other bituminous coals, however, may be burned with 350-F air, and with anthracite and coke breeze air may be as hot as 450 F. High air temperatures accelerates ignition, giving more flexibility with changing loads, and improve over-all boiler efficiency.

The disadvantages of high air temperatures show as warping and overheating of grates. Larger specific volumes of high-temperature air increase its velocity through grate openings, which may lift the finer coal particles off the grate as carry-over.

Example 5-1. A steam generator with maximum continuous steam rating of 200,000 lb per hr will be fired with bituminous coal having 10 per cent moisture and 10 per cent ash and a higher heating value (HHV) of 13,200 Btu per lb. If it takes 1000 Btu to evaporate 1 lb of feedwater entering the boiler and superheat it to final temperature and if the over-all steam-generator efficiency is 80 per cent, find: (*a*) the hourly coal supply in tons at maximum rating; (*b*) the grate area needed; (*c*) the grate length if building dimensions limit grate width to 20 ft.

Solution

1. Total heat absorbed by water and steam $= 200{,}000 \times 1{,}000 = 200 \times 10^6$ Btu per hr

2. Fuel heat release $= \dfrac{200 \times 10^6}{0.80} = 250 \times 10^6$ Btu per hr

3. Coal supply $= \dfrac{250 \times 10^6}{13{,}200 \times 2{,}000} = 9.48$ tons per hr

4. Grate area $= \dfrac{250 \times 10^6}{500{,}000} = 500$ sq ft

5. Grate length $= {}^{500}/_{20} = 25$ ft

Traveling-grate stokers may be driven by variable-speed electric motors or by constant-speed motors through hydraulic- or electric-type couplings.

High-ratio gear reductions must be used to get the speed down to the few feet per hour of the grate travel. The hopper coal gate usually may be regulated by a handwheel through a worm-and-gear arrangement.

Stoker-drive motors vary from about 1 hp for a 100-sq ft grate to 3 hp for a 400-sq ft stoker. The energy consumption for grate drive is negligible compared with the heat released by the fuel. Severe heating of the stoker requires regular repairs as grate parts need replacements. Maintenance costs vary from about 1 to 4 cents per ton of coal burned on the grate.

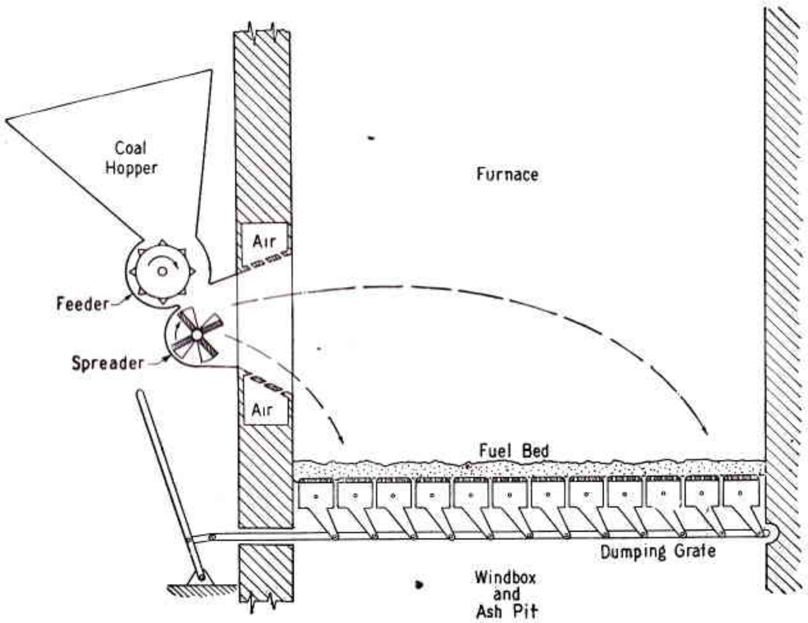


FIG. 5-4. Spreader stoker hurls coal particles into furnace; light coal particles burn in suspension; heavy particles fall on grate and burn. Slowly rotating feeder measures out steady stream of coal falling in front of fast-turning spreader.

5-5. Spreader Stokers. These stokers use the overfeed principle of fuel burning. Figure 5-4 shows the elements of a spreader stoker feeding coal onto a stationary grate. The rotating feeder, consisting of short blades on the surface of a drum, takes coal out of the bottom of the hopper and drops it in a continuous stream to the spreader below. The speed of the feeder varies directly with steam output of the boiler.

The spreader, or distributor, consists of a shaft carrying paddles alternately inclined to the left and right of the furnace center line. The paddles of the rapidly rotating spreader hit the particles of the steady coal stream falling in front of them and hurl them into the furnace. The angularity of the paddles gives uniform distribution of the coal in the furnace from left to right. The distribution along the depth of the furnace depends on the rotating speed of the distributor, which can be varied, and on the size consist of the coal. If too large a proportion of the coal runs to be one size the coal tends to concentrate in one area of the grate. Sizes should be uniformly mixed from fines to about ¾ in. for best results. The spreader speed must be varied as the size consist of the coal changes to maintain uniform distribution on the grate.

Other methods may be used to measure and convey the coal from the hopper to the distributor, such as: (1) a continuous belt and gate under the coal hopper; (2) multiple cylinders and pistons under the coal hopper, the coal dropping into the cylinders from the hopper and the piston pushing the coal out of the cylinder to fall before the distributor; and (3) a worm-screw feed from the bottom of the coal hopper.

Spreaders may be either overthrow as in Fig. 5-4 or underthrow. Fine particles of the coal thrown into the furnace burn in suspension. Part of the volatile matter of the larger coal pieces distills off in the furnace heat before the coal reaches the grate. The total burning process combines suspension and overfeed combustion. The fuel bed on the grate usually is about 3 to 4 in. thick.

Figure 5-4 shows a dumping type of stationary grate; it consists of perforated cast-iron bars about 3 ft long and 6 in. wide with a depth of about 6 in. Each bar pivots parallel to the 3-ft axis. Bars placed side by side with their 3-ft lengths adjacent to each other make up a grate section. Links underneath the grate connect all the bars to a lever near the spreader stoker. Moving the lever back and forth makes the bars rock about their pivots. This makes the ash on the bottom of the fire fall through to the ashpit underneath the grate. Several grate sections with individual dumping levers make up a complete grate. This allows cleaning one part of a fuel bed at a time with only a small drop in steam output.

Larger steam generators may have two or more spreader stokers side by side in the furnace front wall. Stationary grates usually serve boilers up to a maximum of about 75,000 lb per hr capacity. For larger capacities up to about 350,000 lb per hr a traveling grate (Fig. 5-5) will be used. In this design the grate travels from back to front of the furnace, discharging the ash into a pit underneath the stoker. In other arrangements the grate may travel from front to back with the ashpit at the rear.

Except for the larger boilers the wind box under the grate will be compartmented only for the length of the furnace, so that air can be shut off while cleaning a section of the fire. In large furnaces with traveling grates the wind box may be compartmented along the length of the stoker to compensate for fuel-bed thinning.

Heat-release rates on stationary grates may run as high as 450,000 Btu per sq ft-hr; on traveling grates the rate may be doubled to 900,000 Btu per sq ft-hr. Since part of the fuel burns in suspension, the burning rate related to furnace volume plays an important part. In refractory furnaces heat-liberation maximum rates run from 20,000 to 35,000 Btu per cu ft-hr. In water-cooled furnaces this rate may be pushed up to 45,000 Btu per cu ft-hr.

Because of the thin fuel bed and suspension burning spreader stokers tend to suffer a fuel loss in cinders up the stack. This may run from about 1 to 4 per cent. Arrangements are usually made to return some of the cinders, trapped in the boiler passes, to the furnace for complete burning.

Mixing of the furnace gases and air may be done by jetting secondary air into the furnace about 3 ft above the grate. Some designs may use steam jets for the purpose and force all combustion air through the fuel bed. These jets also help to reduce the cinder loss and smoke.

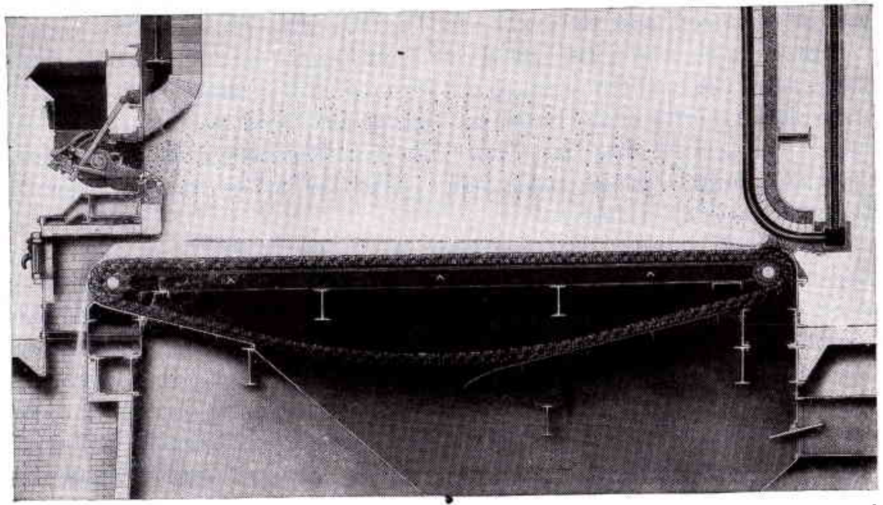


Fig. 5-5. Spreader stoker scatters coal evenly on chain grate moving slowly toward stoker end of furnace. At end of travel ashes drop off stoker into pit below.

5-6. Underfeed Stokers. Figures 5-6 and 5-7 show essential elements of one type of *multiple-retort underfeed stoker*. The stoker consists of a series of sloping parallel troughs formed by tuyere stacks. These troughs are called *retorts*. Under the coal hopper at the head end of the retorts, *feeding rams* reciprocate back and forth. With the ram in the outer position coal from the hopper falls into the space vacated by the ram. On the inward stroke the ram forces the coal into the retort. Ideally, raw coal completely fills the retort to a level above the tuyeres.

Secondary, or *distributing, rams*, forming the floor of the retorts, control the height and profile of the fuel bed (Fig. 5-7). These rams oscillate parallel to the retort axis; the length of their strokes can be varied as

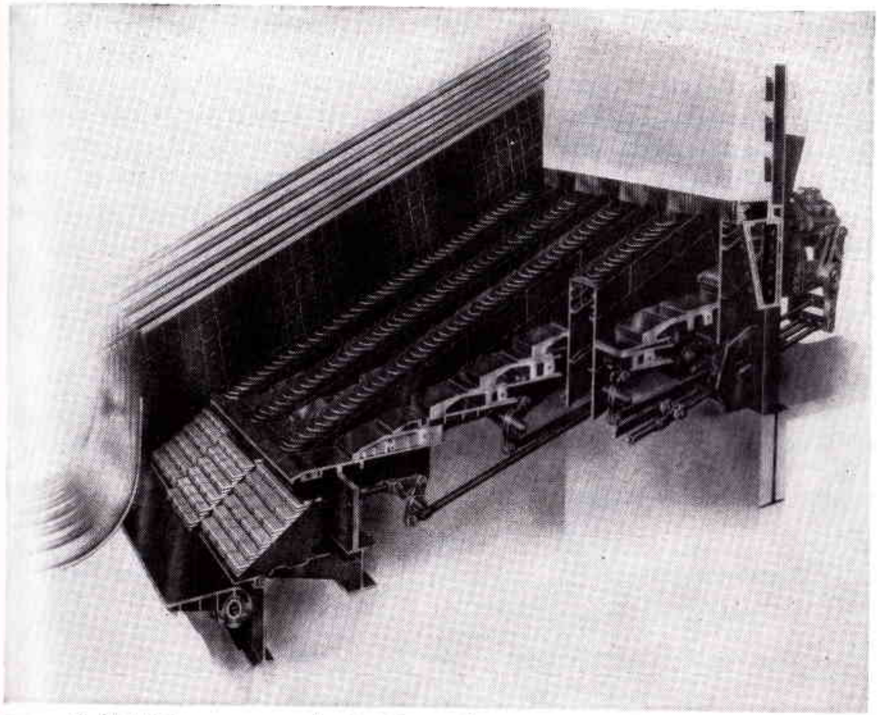


Fig. 5-6. Multiple-retort underfeed stoker section shows, from left to right, ash dump plate, reciprocating extension grate; distributing rams, and section through tuyere stack. (*Courtesy of American Engineering Company.*)

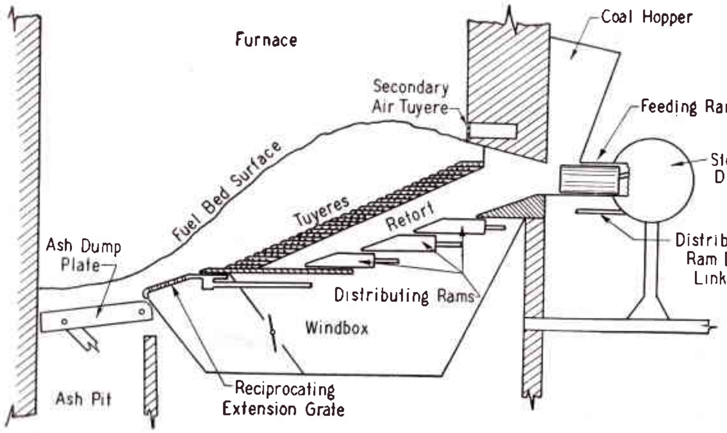
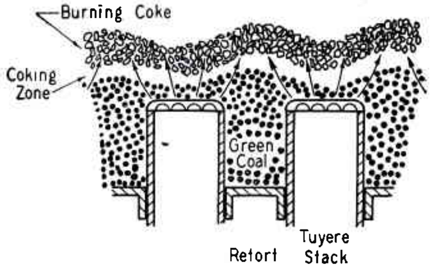
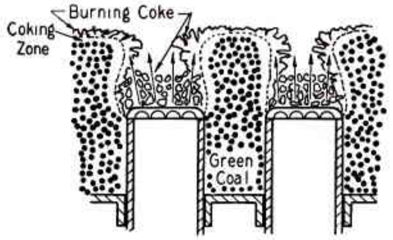


Fig. 5-7. Coal from hopper drops in front of reciprocating ram to be pushed into retort bottom. Distributing rams force coal upward and to the left as fuel bed slowly progresses to extension grate and ash dump plate.

needed. They slowly move the entire fuel bed down the length of the stoker. At the rear of the stoker the partly burned fuel bed moves onto an *extension grate* arranged in sections. These sections also oscillate parallel to the fuel-bed movement. The sharp slope of the stoker aids in moving the fuel bed. Fuel-bed movement keeps it slightly agitated to break up clinker formation. From the extension grate the ash moves onto the ash dump plate. Tilting the dump plate at long intervals deposits the ash in the ashpit below.



(a) Ideal fuel bed section



(b) Actual fuel bed section

Fig. 5-8. (*a*) In ideal fuel-bed section on underfeed stokers ignition plane should stay entirely above tuyeres. (*b*) In actual fuel bed ignition plane extends from tuyeres to cap each retort. Burning coke breaks off lanes above retort and falls onto tuyeres to complete burning.

Primary air from the wind box underneath the stoker enters the fuel bed through holes in the vertical sides of the tuyeres. The extension grate carries a much thinner fuel bed and so must have a lower air pressure under it. A controlling air damper regulates the air entering from the main wind box into the extension-grate wind box (Fig. 5-7).

Figure 5-8*a* shows a section of an ideal fuel bed on an underfeed stoker. Green coal fills the retorts to a level above the tuyeres. The ignition plane is a short distance above the tuyeres. Primary air supplied through the tuyeres should diffuse throughout the upper region of the green coal before passing through the ignition plane and the distillation or coking zone. The air and volatile matter then pass through the incandescent zone and finally through the layer of incandescent ash into the furnace above.

Research reveals, however, that the fuel bed acts as in Fig. 5-8*b*. Cool green coal extends well above the tuyere level in the retorts. The ignition surface and coking zone form an inverted U, capping the green coal in the retorts. A thin layer of coke burns at a very slow rate at the horizontal surface above the retorts.

Rapidly burning coke fills the lanes above the tuyeres. Since the air barely penetrates from the tuyeres to the spaces above the retorts, it flows directly to the spaces above the tuyeres. The coke coating the vertical walls of the green coal in the retorts swells and cracks. This makes it fall into the lanes above the tuyeres, where it burns at a fast rate.

The ash eventually works its way down the retort lanes, owing to agitation from the air flow, and finally deposits on the ash dump plate. The mechanics of the combustion process follow neither pure overfeed nor pure underfeed burning on the underfeed stoker.

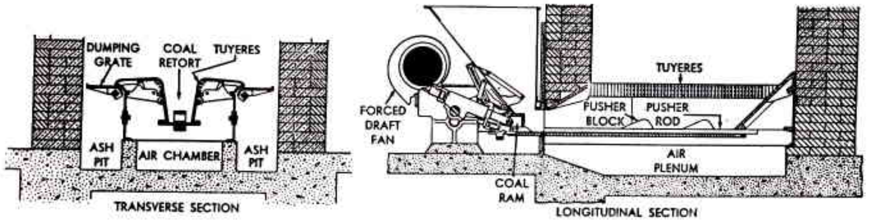


Fig. 5-9. Single-retort underfeed stoker has horizontal retort bottom. Reciprocating ram, or screw feeder, forces coal into bottom of retort. Distributing rams help force coal upward to spill over tuyeres so that burning fuel bed moves sideways to ash dump plates at sides of stoker.

Multiple underfeed stokers have been used in steam generators up to 500,000 lb per hr capacity. The advantages of pulverized fuel in large boilers, however, have caused this method to supersede underfeed stokers for central-station service. Underfeed stokers still find a place in medium-sized and small boilers used for industrial service.

Single-retort underfeed stokers serve smaller boilers, their size ranging from 10 to about 130 sq ft of projected area. These correspond to coal-burning capacities of about 100 to 7,000 lb per hr and steaming capacities of 700 to 60,000 lb per hr. Figure 5-9 shows one form of single-retort underfeed stoker. The variety of designs available usually have horizontal retorts. Either a worm or a ram forces coal into the retort from the bottom of the hopper. Adjustable reciprocating feed rams make up the bottom of the retort; these aid in distributing the coal along the length of the retort. Coal forced in along the length of the retort spills up over the tuyeres on the sides of the retort. The action of the rams keeps the coal moving so that fluid ash will not mat the fuel bed. Additional agitation comes from mechanically moving the grate on each side of the retort. This motion gradually moves the fuel bed sideways from the retort onto the dump plates at the sides. Tilting the dump plates drops the ash into the pits below.

PROBLEMS

5-1. A steam plant has a 2,000-kw turbine that uses 12 lb of steam per kilowatt-hour at 400 psia and 500 F. Steam-generator efficiency is 75.0 per cent when burning coal with a HHV of 13,610 Btu per lb and when supplied with 280-F feedwater. The underfeed stoker serving the furnace needs 0.5 hp per 1,000 lb of coal burned per hour.

Calculate the drive power needed by the stoker as a per cent of the turbine output at full load.

5-2. A spreader stoker with traveling grate can burn 40 lb per sq ft-hr of 13,200-Btu-per-lb coal. In a refractory-walled furnace the designer determines he should limit the heat release to 20,000 Btu per cu ft-hr to avoid high maintenance. If the grate dimensions are 15 by 20 ft and the projected ashpit area is 40 sq ft, how tall should the furnace be?

5-3. A traveling-grate stoker burns coal with a HHV of 12,600 Btu per lb costing 30 cents per million Btu. If it costs 3 cents per ton of coal burned to repair the grate, what percentage of the total fuel-cost does this represent?

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CHAPTER 6

SUSPENSION FIRING

6-1. Pulverized Coal. Burning powdered (pulverized) coal for generating steam was first applied commercially in the early 1920s. Since then it has become almost universal in central stations using coal for fuel. This method first grinds coal to dustlike size and then carries the powdered coal in a stream of air into the furnace.

High-temperature flames in the furnace heat the entering coal particles to distill off the volatile matter. The volatile gases mix with the oxygen of the air, further heating ignites them, and they burn quickly. Distillation reduces the coal particles to minute, spongelike masses of fixed carbon and ash.

Heating the carbon raises it to ignition temperature, and it reacts with the oxygen of the hot air in the furnace to release heat energy. This forms a blanket of combustion products on the carbon particles. To expose the unburned carbon underneath to fresh supplies of oxygen, the blanket of gaseous products must be stripped off by turbulent mixing of the carbon particles and air.

As explained in Sec. 5-2, proper fuel burning requires supplying correct proportion of air, mixing of fuel and air, high temperature, and adequate time to complete combustion reactions. The ash remaining after completing combustion reactions falls to the furnace bottom, or floats in the combustion gases to the boiler flue-gas outlet, or deposits on the boiler heating surfaces.

Pulverized-coal furnaces must withstand the high temperatures and be properly proportioned to hold the volume of fuel, air, and gases and allow enough time for complete burning. Modern central-station boiler furnaces have water-cooled walls that form part of the steam-generating heat-absorbing surfaces.

The success of pulverized-coal burning stems from the greater surface exposed by breaking a given mass of coal into smaller pieces. Figure 6-1 shows the relation between surface per unit volume of spheres (or cubes) with diameter (or edge length). For example, at 0.1 in. diameter a sphere has 60 sq in. surface per cubic inch volume; at 0.01 in. it has 600 sq in. per cu in.; and at 0.001 in. it has 6,000 sq in. per cu in. The steeply rising curve shows the advantage of fine pulverization.

Greater surface per unit mass of coal allows faster combustion reactions because more carbon becomes exposed to heat and oxygen. This reduces the excess air needed to ensure complete combustion. In turn this reduces the dry-gas loss and raises the over-all steam-generating efficiency. Added costs of coal-grinding equipment and grinding energy partly offset these advantages.

Pulverized-coal firing advantages include: (1) low excess-air needs; (2) lower fan power; (3) higher boiler efficiency; (4) ability to burn wide variety of coals; (5) fast response to load changes; (6) low banking losses; (7) ease of burning alternately with, or in combination with, gas and oil; and (8) ability to release large amounts of heat, making it possible to generate about 4,000,000 lb steam per hour in one boiler.

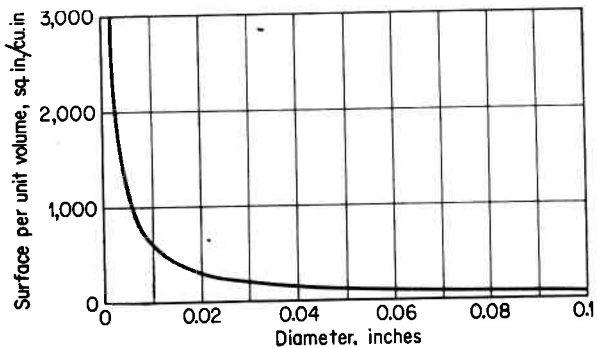


FIG. 6-1. Reducing the size of particles exposes more surface per unit volume. This relation applies to spheres or cubes; for the latter substitute edge length for diameter.

The disadvantages of pulverized coal include: (1) added investment in coal-preparation equipment, (2) added power needed for pulverizing coal, (3) investment in stack fly-ash removal equipment, and (4) large-volume furnaces needed to withstand high gas temperatures. These disadvantages offset some of the advantages cited, but the net gain has led to the wide use of pulverized-coal firing.

6-2. Pulverizing Systems. The *bin*, or *central, system* was the first arrangement used in pulverizing coal for steam generation. Figure 6-2 indicates the sequence of equipment needed and features a closed storage bin for the powdered coal. The advantages of the bin system include the ability to grind coal at a steady rate regardless of boiler load. Less pulverizer (mill) capacity need be installed for a given boiler capacity, and the mills can be run steadily at their most efficient rating. Occasional mill outage does not affect coal feed to the furnace.

Additional equipment needed for separating conveying air from the pulverized coal before storing in the bin, and the conveying system itself, offset the economy of lower mill capacity. In addition the nuisance of residual coal dust carried to the atmosphere by the air separated in the cyclone collector and the hazard of storing pulverized fuel create operating problems.

While several bin systems still run in some central stations, practically all modern plants use the *unit*, or *direct-fired*, system. Figure 6-2 also shows the arrangement of the direct-fired system, which eliminates the intermediate storage bin. Pulverized coal from the mill discharges directly to the furnace. We shall confine our attention to the direct-fired system.

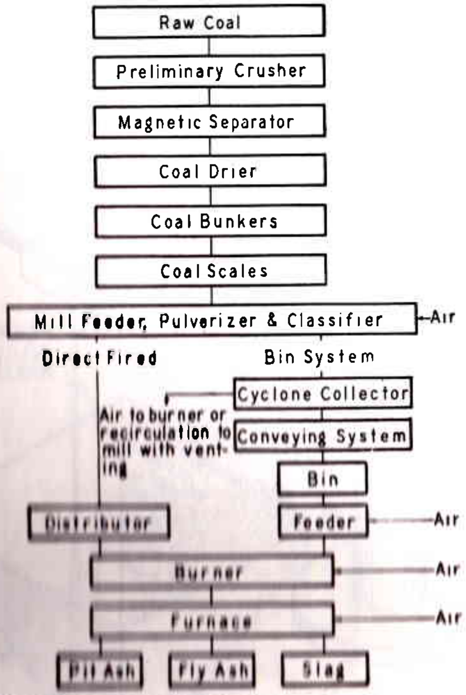


FIG. 6-2. Coal handling for a pulverized-fuel plant involves a sequence of equipment that sizes, dries, stores, measures, mills, sorts, and transports the coal from plant entry to boiler furnace.

6-3. Preliminary Equipment. Coal received by a plant may vary widely in sizes of individual pieces. For efficient grinding, coal should not exceed certain dimensions. Many plants pass the raw coal through a preliminary crusher to reduce the size to allowable limit. This crushing usually takes place as an operation in removing the coal from barge, train, or truck and transporting it to the coal storage pile or bunker in the plant. In such handling the coal often moves by conveyer belt. At the conveyer discharges, magnets remove pyrites and "tramp iron" that mixed with the coal on its way from the mine to plant.

I. E.

In a few installations, especially the older bin systems, the coal may pass through a dryer to remove excess moisture. But the present practice of using heated air in modern mills has made preliminary drying obsolete except for unusually wet coals.

6-4. Coal Pulverizers. Mills reduce coal to powder form by three actions: (1) impact, (2) attrition, and (3) crushing. Most mills use all three actions in varying degrees depending on design. In impact, hammers break up the coal into smaller pieces; in attrition, the coal pieces rub against each other or against metal surfaces to disintegrate; in crushing, coal caught between metal rolling surfaces breaks down to dust.

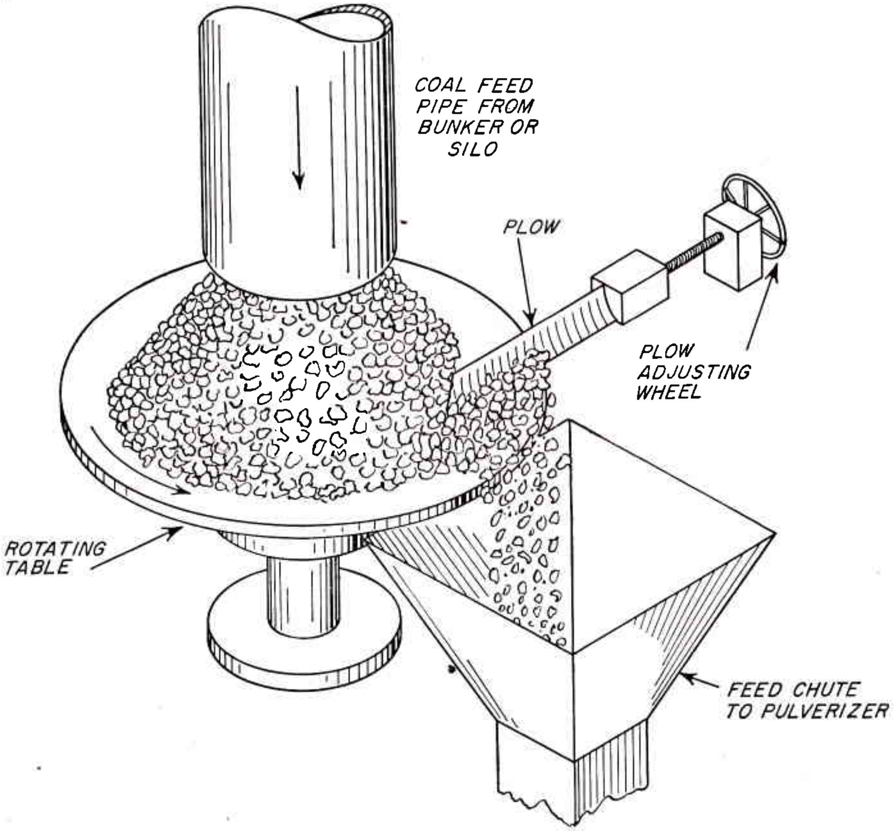


FIG. 6-3. In table feeder coal from spout piles up on rotating table. Adjustable stationary blade scrapes coal from rim of coal pile into chute below, to enter pulverizer.

Mills must have reliable methods of feeding coal to the grinding section and removing the powdered product. For direct firing the coal must be fed at a rate varying with the steam output of the boiler. Figure 6-3 shows the elements of a disk, or plate, feeder. Coal from a bin above falls onto the slowly rotating table through the downspout. The distance between the downspout and table is fixed so the coal does not fall off the table but backs up into the spout. The plow cuts away the outer edge of the coal pile on the table, making a measured amount fall into the feed chute to the pulverizer. Adjusting the plow position and table rotation controls the rate of coal flow entering the mill.

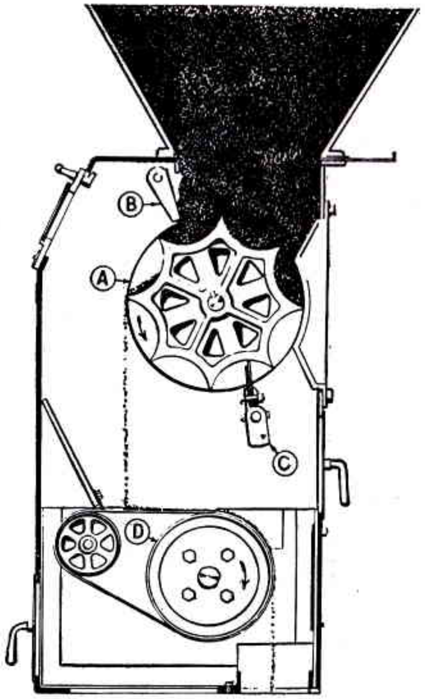


FIG. 6-4. In drum feeder coal from above fills pockets of rotating drum to be metered. Pockets discharge coal onto belt below fitted with magnetic separator in wheel *D*. Belt discharges coal to pulverizer below. Wiper *C* removes coal sticking to drum pockets.

Figure 6-4 shows a drum-type coal feeder. Coal from a bin above fills the pockets of the drum as it revolves slowly. The spring-loaded leveling apron at the left side of the hopper outlet measures out equal coal amounts into each pocket. A variable-speed transmission varies the drum rotation to control the coal-feed rate. A hinged wiper at the bottom of the drum removes wet coal that adheres to the drum pockets. From the pockets the coal falls onto the conveyor belt to be discharged to the pulverizer below.

6-5. Ball, or Tube, Mill. Figure 6-5 shows a ball, or tube, mill. This has a slowly revolving horizontal cylinder turning at about 20 rpm. About half its volume holds a mixture of steel balls and coal that rides up the rising side of the cylinder. The upper edge of the mixture avalanches down the inclined side as it rises too high. This continuous tumbling action makes the heavy balls slam against the coal pieces, reducing them to powder by combined impact, attrition, and crushing actions.

Raw coal from the table feeder, upper right, drops to the ends of the cylinder's hollow annular shaft, where the worms push it into the drum to mix with the steel balls. Heated air enters the cylinder, or drum, through the hollow tubes at each end. The swirling air picks up the coal dust and leaves via the annular spaces through which the raw coal enters the cylinder. The fast-moving coal-air mixture enters the classifiers at each end of the shaft. Here it makes several sharp changes in direction that throw out oversized coal particles. These *tailings* fall, to reenter the drum with the raw coal.

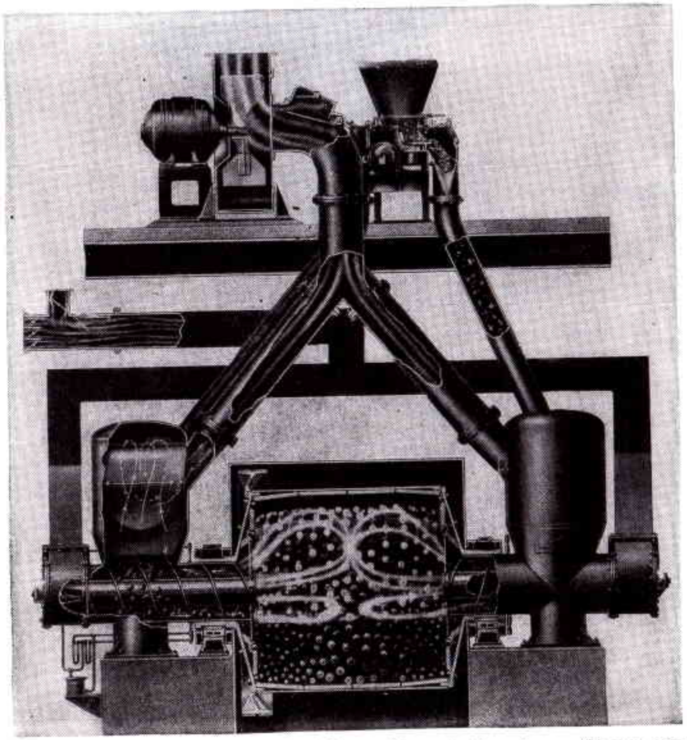


FIG. 6-5. Heavy steel balls tumbled with coal in rotating drum (lower center) grind coal to powder. Raw coal from feeder (upper right) drops to shaft ends. Worms push coal into drum. Heated air enters drum through center ducts on each side, picks up powdered coal, and exits through annular space in shafts. Coal-air mix flows through classifiers at each shaft end to reject oversize pieces to drum for more grinding. Exhauster fan (above left) passes mix to burners. *(Courtesy of Foster Wheeler Corporation.)*

The exhauster fan, above left, induces the air flow through the mill and passes the dust-laden air on to the distributing ducts for delivery to the burners in the furnace wall. The coal output of the mill depends on the rate of air flow. Since a constant-speed motor drives the fan, a damper at the fan inlet controls air-coal flow to the burners. Automatic control varies the coal-feeder rate with the rate of air flow.

The coal being milled in the drum provides a reserve of about 8 min should the raw coal feed be accidentally interrupted. The mill produces 70 to 75 per cent of coal dust passing through a 200-mesh sieve when grinding bituminous coals. About 0.1 to 0.25 lb of steel balls wear away for each ton of bituminous coal milled. Soft coals consume about 12 kwhr per ton and medium-hard bituminous up to 17 kwhr per ton to drive the mill. The feeder needs up to 1 kwhr per ton and the exhauster fan up to 7 kwhr per ton of coal milled.

6-6. Ball-and-race Mill. The ball-and-race mill (Fig. 6-6) crushes coal between two moving surfaces, balls and races. An upper stationary race and a lower rotating race, driven by a worm and gear, hold the balls between them. Coal enters the mill through the rotating-table feeder, at the upper right, to fall on the inner side of the races. The moving balls and races catch coal between them to crush it to a powder. Springs hold down the upper stationary race and adjust the force needed for crushing.

Hot air supplied by a forced-draft fan enters the mill through the annular space surrounding the races. As it flows between the balls and races, it picks up the coal dust and enters the classifier above. The fixed vanes make the entering air take on a cyclonic flow which throws the oversized coal particles to the wall of the classifier. From here they slide down for further grinding in the mill. Dust of proper sizes leaves with the air through the top of the classifier on its way to the burners.

While the tube mill (Fig. 6-5) runs under negative pressure because the fan is at the mill outlet, the mill in Fig. 6-6 runs under positive pressure because the fan is at the inlet. The fan handles air only, which avoids eroding the fan blades by coal particles passing through.

The lower race turns at about 220 to 280 rpm. The mill can handle coals with as much as 20 per cent moisture. The air-flow rate fixes the amount of coal dust removed from the mill with the coal-feeder speed responding through automatic controls. Mill, feeder, and fan need up to 14 kwhr per ton of coal pulverized.

6-7. Bowl Mill. The bowl mill shown in Fig. 6-7 has a grinding ring carried on the sloping side of the rotating bowl. The bowl rotation gives the bowl a rim speed of about 1,200 ft per min. The spring-loaded stationary rollers just clear the grinding ring.

A rotary feeder, not shown, discharges coal into the turning bowl.

Centrifugal force throws the coal to the bowl rim and up the sloping sides, to be crushed under the rollers. From the bowl rim the coal enters the surrounding annular passage. Deflectors above the bowl bounce the large particles back into the bowl.

Hot air enters the mill at the left side of the mill bowl and flows upward and then horizontally across the bowl top, to pick up the coal fines. It enters the classifier, taking on a cyclonic flow that throws the oversized particles outward, to be returned to the bowl through the central spout for more grinding.

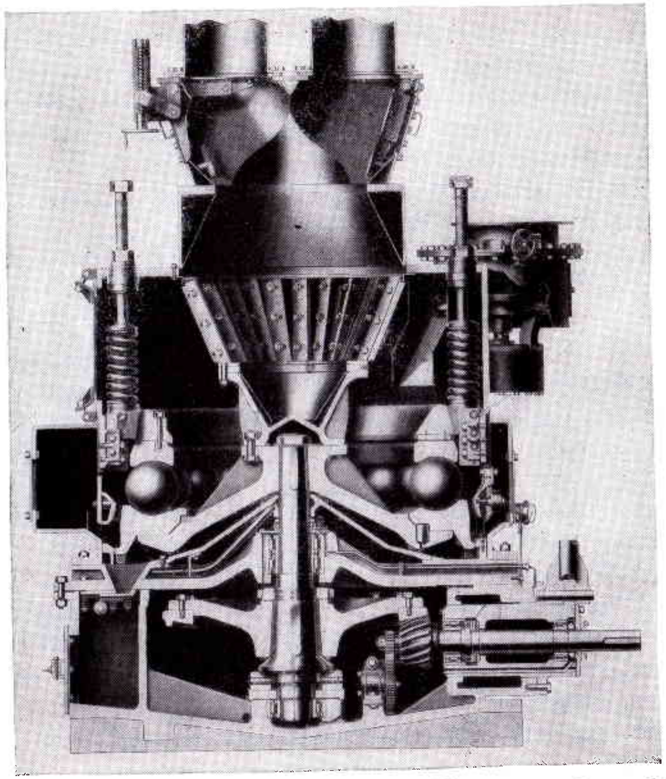


FIG. 6-6. Ball-and-race mill has lower race turned by worm-and-gear drive. Balls turn between lower race and upper spring-loaded stationary race. Coal enters from feeder, upper right, to drop inside races. Air from annular duct, surrounding pulverizer, enters from beneath lower race to pick up coal crushed between balls and races. Coal-air mix enters classifier in center above vertical shaft. Tangential entry between blades throws oversized coal pieces to sides, to slide back to mill. Coal-air mix leaves through top on way to burners. (*Courtesy of The Babcock & Wilcox Company.*)

Pyrites and tramp iron pass up to and over the bowl rim by centrifugal force, to fall to the mill bottom. Arms on the bowl bottom sweep them around, to be discharged through the lower right spout.

The exhauster fan, lower left, takes the dust-laden air from the classifier and passes it on to the burners. Mill and feeder use about 10 kwhr per ton of coal ground and the exhauster fan about 5 kwhr per ton.

6-8. Pulverizer Performance. The power consumption of a mill should be low to minimize operating costs. This main factor of performance depends on: (1) grindability of the coal, (2) surface moisture on the coal, and (3) fineness of the grind needed.

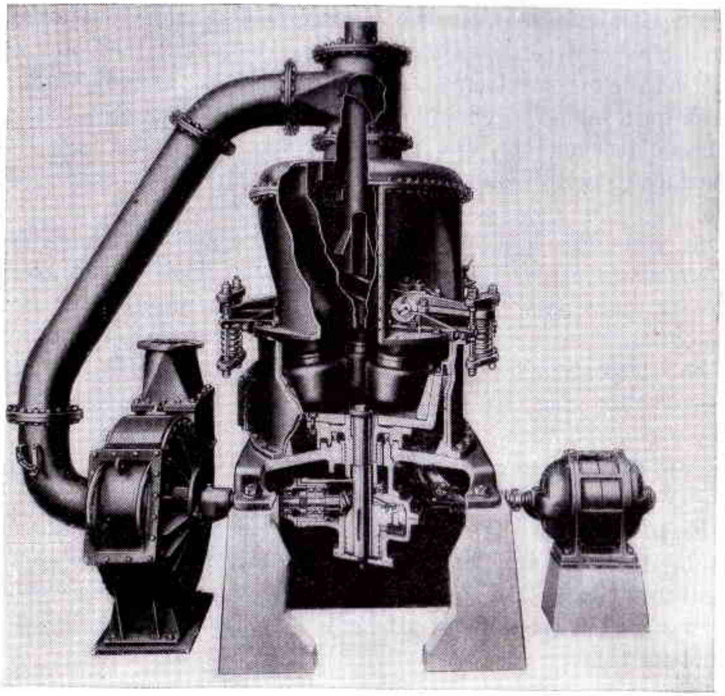


FIG. 6-7. Bowl mill turned by worm-and-gear drive receives coal from drum feeder at upper right, not shown. Coal caught between bowl side and stationary spring-loaded rollers is pulverized and thrown to bowl rim. Hot air enters through duct, left of bowl, passes up outer side of bowl, picks up powdered coal, and enters classifier above. Oversized pieces fall back to bowl through center spout. Exhauster fan pulls coal-air mix out of classifier and passes it on to burners. (*Courtesy of Combustion Engineering Company, Inc.*)

The method of measuring coal grindability that is most often used is the Hardgrove index. A low index number designates coals hard and difficult to grind, while a high index number applies to coals soft and easy to grind. The method depends on grinding properly prepared and measured coal samples in a small laboratory mill for 60 revolutions and then measuring the amount that passes through a 200-mesh sieve.

Hardgrove grindabilities vary from less than 26 to over 112. The capacity of a mill varies almost directly with grindability, rising not quite so rapidly with the higher indexes. The amount of coal that a mill can handle depends entirely on the grindability of the coal, an uncontrollable factor, and one that a plant designer must carefully investigate to ensure that a steam generator can fully develop its rated capacity when burning the given coal.

The fineness of the powdered coal needed for successful burning depends largely on the ratio of volatile matter (VM) to fixed carbon. For anthracite (low VM) about 85 per cent of the powdered coal should pass through a 200-mesh sieve, with less than 2 per cent being retained on a 50-mesh screen. At the other extreme only about 60 per cent need pass through a 200-mesh for a high-VM coal like lignite. Since the carbon burns slower than the volatile matter, low-VM coals must be ground finer to expose more surface to permit burning more coal in a given residence time in the furnace.

The finer the grind, the greater the power input needed. Overgrinding is expensive in energy and maintenance incurred by the extra wear. This factor must be watched to achieve efficient plant operation. For example, with 70 per cent through 200-mesh, power and maintenance cost would be relatively 100 per cent. But at 80 per cent through 200-mesh cost rises to 135 per cent, and at 90 per cent through 200-mesh cost rockets to 245 per cent.

Moisture makes fine coal particles adhere to each other, defeating the purpose of grinding to expose more surface. Heated primary air evaporates this moisture in the mill and carries the vapor to the furnace with the coal. Inlet-air temperature must be regulated to keep the exit temperature at least 25 F higher than the dew point of the exit air; this usually ranges between 130 and 160 F. With some coals this requires a mill inlet-air temperature up to 600 F.

6-9. Burners. Primary air that carries the powdered coal from the mill to the furnace is only about 20 per cent of the total air needed for combustion. Before the coal enters the furnace, it must be mixed with additional air, known as *secondary air*, in burners mounted in the furnace wall.

In addition to the prime function of mixing, burners must also maintain stable ignition of the fuel-air mix and control the flame shape and travel in the furnace. Ignition depends on the rate of flame propagation. To prevent flashback into the burner, the coal-air mixture must move away from the burner at a rate equal to flame-front travel. Too much secondary air can cool the mixture and prevent its heating to ignition temperature.

Turbulent burners (Fig. 6-8) give a rotary motion to the coal-air mixture in a central nozzle and also to the secondary air that flows through

the annular space surrounding the nozzle. This mixes the two streams, as they enter the furnace at the left, to distribute the coal dust uniformly throughout the total air available for combustion.

The multitip burner (Fig. 6-9) has 16 tips, or nozzles, fed by a duct from the mill. This aids in uniformly distributing the primary-air and coal streams entering the furnace. The secondary air enters the furnace in close proximity to the coal through separate openings. Mixing takes place in the furnace, and not the burner, in this type.

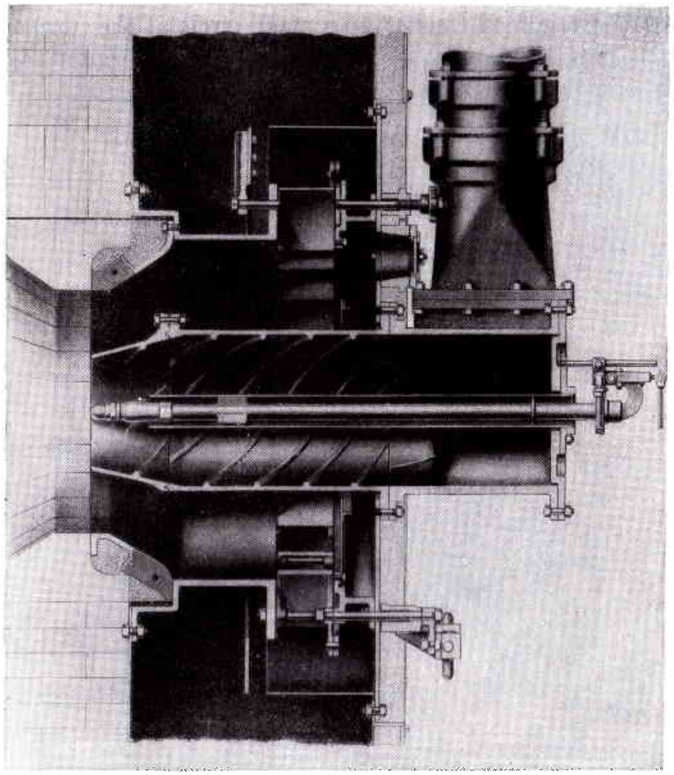


Fig. 6-8. Turbulent burner for coal and oil firing has side entrance to center annular tube so that air travels spiral path into furnace, left. Oil sprays through mechanical atomizer nozzle, left center. Secondary air enters through dampers surrounding burner.

This vertical arrangement projects the flame downward, but before leaving the furnace the flame usually makes a U turn upward. This gives it longer travel and time to complete the combustion reaction. Slag-tap furnaces often use this type of burner because the flame being just a short distance above the furnace bottom keeps the ash molten so that it can flow out to a slag tank below.

Tilting tangential-fired burners (Fig. 6-10) aid in controlling the superheat of the steam leaving the steam generator. One burner assembly stands in each of the four corners of the furnace at about midheight (Fig. 6-11). Pipes from the mills feed the coal nozzles aimed into the furnace. Secondary air flows through the annular areas surrounding the nozzles. The coal nozzles can be tilted to vary the flame zone vertically over a range of about 12 ft.

The burners are aimed tangent to a small circle at the furnace center. The streams impinge on each other near the furnace center to promote adequate mixing and turbulence.

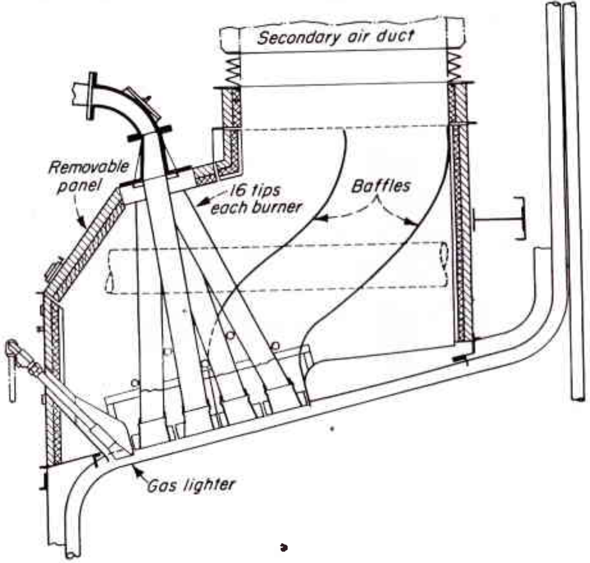


FIG. 6-9. Vertical powdered-coal burners with 16 tips direct coal-air mix toward bottom of furnace.

The oil-burning nozzles in Figs. 6-8 and 6-10 are used to ignite the coal during start-up of the steam generator. An oil flame produces a temperature high enough to ignite the coal. Once the coal flame is established, the oil flame is extinguished.

Single coal burners have capacities up to 150 million Btu per hr heat release at full load. This equals about 6 tons of coal per hour. Central-station boilers have several pulverizers, each feeding several burners. Some burners have coal nozzles, each fed from the different pulverizers serving the boiler. The total heat release in the boiler furnace can be varied by changing coal and air flow through all the pulverizers and their connected burners. Most pulverizers have a turndown ratio of about 1:3, the minimum heat release or fuel-air flow being one-third the full-load flow. To achieve a greater over-all turndown ratio an entire

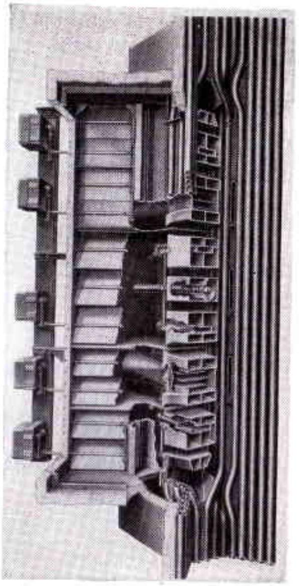
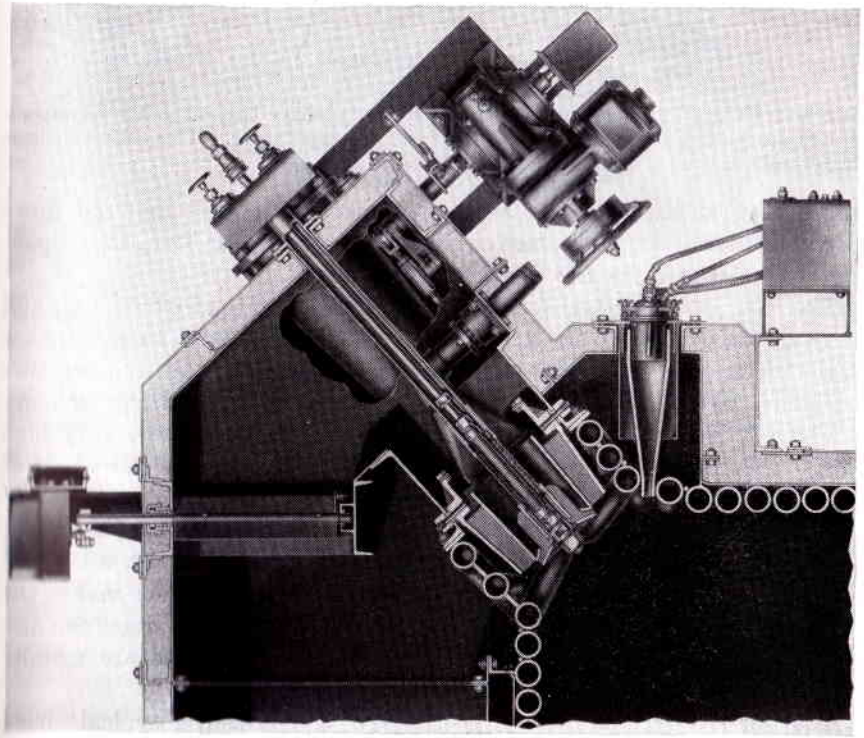


FIG. 6-10. Corner-mounted pulverized-coal burners have tilting nozzles to vary height of flame zone in furnace. *(Courtesy of Combustion Engineering Company, Inc.)*



pulverizer and its connected burner coal nozzles can be shut down to get lower boiler outputs.

Air flow must not be reduced below a velocity of about 3,000 ft per min through the pulverizer-burner system to ensure keeping the powdered coal in suspension. The maximum flow rates range up to 5,500 ft per min; fan power requirements to overcome pressure drop in the coal-air piping set the upper limit from an economy standpoint.

6-10. Cyclone Furnace. The cyclone furnace (Fig. 6-12) was primarily designed to burn the poorer and cheaper grades of coal. These have higher ash content and ash-softening temperatures of less than 2500 F. The furnace is a horizontal cylinder, completely water-cooled, that receives crushed coal (up to ¼ in. maximum size) carried in primary air at the left end.

Tangential entry of the coal throws it to the surface of the cylindrical furnace. Secondary air enters the furnace through tangential ports at the upper edge at high speed. Molten slag coats the cylinder and flows slowly down the curved surface of the cylinder, to drain through the opening at the lower right into the primary furnace.

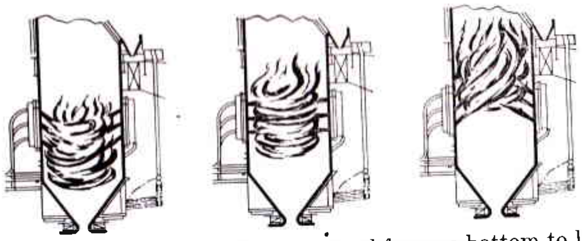


FIG. 6-11. Tilting burners can direct flames toward furnace bottom to lower temperature of gas leaving furnace or toward top to raise temperature. This controls superheater-outlet steam temperature.

The fines in the crushed coal burn in suspension in the turbulent air and gases of the cyclone furnace. The heavier pieces thrown to the surface of the furnace are caught in the molten slag. Since the high-temperature secondary air sweeps over these trapped coal pieces at high velocity (about 350 ft per sec), the combustion reaction proceeds at a very high rate. The furnace develops temperatures of 3000 F and releases heat at rates higher than 500,000 Btu per hr-cu ft.

The combustion gas leaves the cyclone through the throat, at the right, to enter the primary furnace of the boiler. The high heat release in the cyclone furnace (usually more than one serves a boiler) makes the entire unit more compact for a given steam output. Oil and gas can also be burned as alternate fuels in cyclone furnaces. These are usually injected at the secondary-air ports in the furnace.

Coal for the cyclone furnace need not be pulverized; a typical sizing

passes about 50 per cent through a 28-mesh sieve and only 10 per cent through 200-mesh. Cyclone-furnace pressures range up to 40 in. H_2O and eliminate the need for induced-draft fans. The crusher needs less power than a pulverizer. The total auxiliary power for a forced-draft fan and coal crusher runs higher than for a pulverized-coal boiler burning a high-heating-value coal, but less than for one burning a low-heating-value coal.

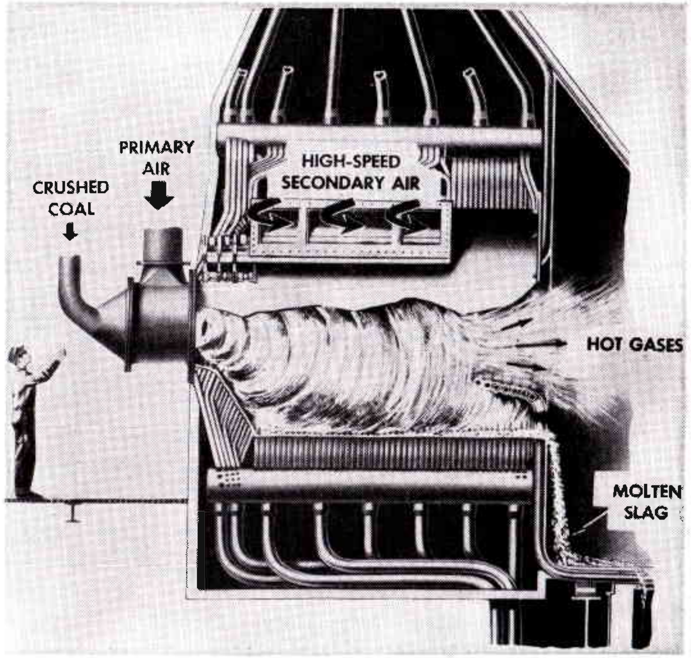


FIG. 6-12. Cyclone furnace throws crushed coal particles to catch in slag flowing down walls of horizontal cylindrical furnace. High-speed secondary air sweeping over trapped coal produces very high rate of combustion reaction. *(Courtesy of The Babcock & Wilcox Company.)*

6-11. Oil-burning Systems. These systems usually fire Bunker-C oil, a residual remaining after the lighter fractions have been removed from crude oil in the refineries. In the typical system shown in Fig. 6-13 oil flows from storage tanks *A*, through strainers *B*, to pumps *C*. These deliver the oil to heaters *D*; then it flows through strainers *E* on its way to the burners *F*. Duplicating this equipment permits cleaning it without shutting down the boiler.

Tanks of concrete or steel store the oil near the plant; capacities may range up to several hundred thousand gallons. Oil tanks must be completely surrounded by a dike that can hold all the oil contents if the tank should rupture.

Oil burners must atomize the oil before injecting it into the furnace. They can do this by (1) heat vaporization or (2) mechanical spraying. Because of the amount of heat needed to vaporize oil the first method is little used in central stations.

Oil must be atomized to expose the maximum possible surface to combustion reactions. Mechanical spraying can be done by: (1) using pressurized air or steam to break up the oil, (2) forcing oil under pressure through small orifices, or (3) tearing oil film into tiny drops by centrifugal force. Steam- and air-atomizing burners are simple and relatively inexpensive. Steam-atomizing burners can handle almost any fuel oil of any viscosity at any temperature. Compressed air is not widely used because of its cost.

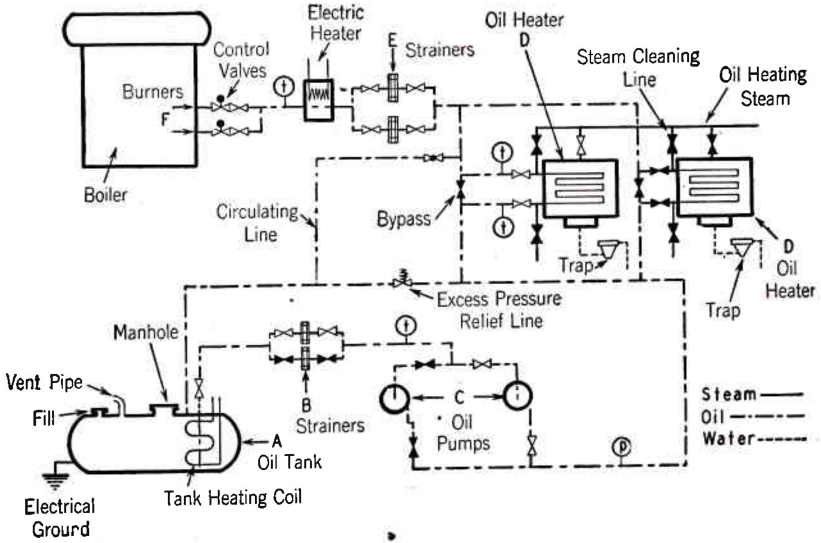


Fig. 6-13. Oil-burning system must store fuel, heat it to reduce viscosity, strain it to remove dirt, pump it to move it, and pressurize it for atomization in burner nozzle.

Figure 6-14 shows an internal-mixing type in which air or steam mixes with the oil before spraying into the furnace. The air or steam picks up oil to force the steam-oil or air-oil mixture through the nozzle.

Figure 6-15 shows an external-mixing type of burner. Oil flows to the tip through the central annular passage. The screw spindle regulates the oil flow. Oil whirls out against a sprayer plate, to break up at right angles to the steam flow (or air flow) coming out behind it on the way into the furnace. The atomizing stream surrounds the oil passage and takes a whirling motion from the guiding vanes in the path.

Steam for atomizing ranges from 1 to 5 per cent of steam generated, averaging about 2 per cent. Steam pressure at the burner must be about 75 to 150 psig and oil pressure only about 10 to 15 psig. Atomization is usually best at one flow rate; so for major load changes on the boiler oil burners are cut into and out of service as needed.

Forcing oil at pressures of 75 to 200 psig through small orifices gives good atomization. Disks placed near the nozzle give the oil a whirling motion before it passes through the orifice and helps form smaller drops. Figure 6-16 shows burner and pumping systems that have turndown ratios of over 1:15. The upper system uses a constant-differential valve and the lower system a constant-differential pump to hold a constant difference in pressure between oil supply and return. This holds a uniform pressure drop across the tangential slots in the burner tip to develop uniform atomization at all oil-flow rates. The pump system has the following advantages: (1) no hot-oil return to storage tank or pump suction, (2) oil enters closed circuit at the same rate it burns, and (3) pump can boost pressure on existing oil-burner systems.

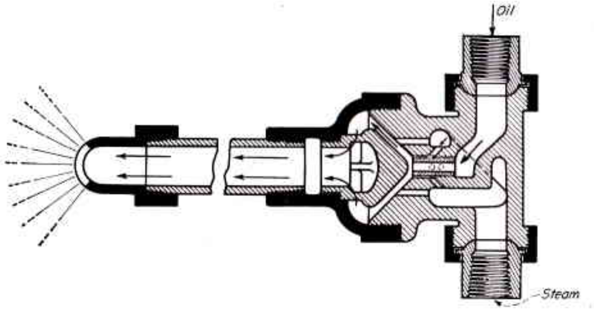


Fig. 6-14. In internal-mixing steam-atomizing burner the oil and steam mix for atomizing oil before entering furnace.

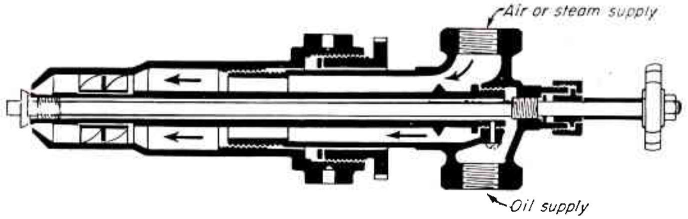


Fig. 6-15. In external-mixing burner oil and air meet at burner tip to atomize the oil prior to burning in furnace.

Figure 6-17 shows a rotary-cup oil burner that turns at about 3,500 rpm. Oil pumped to the interior of the cup splashes on the inside surface. Centrifugal force moves the oil in a thin film to the rim, where it flies off into the air stream discharged by the integral blower. High oil temperatures must be avoided to prevent vaporization. The rotary-cup burner has turndown ratios of 1:16.

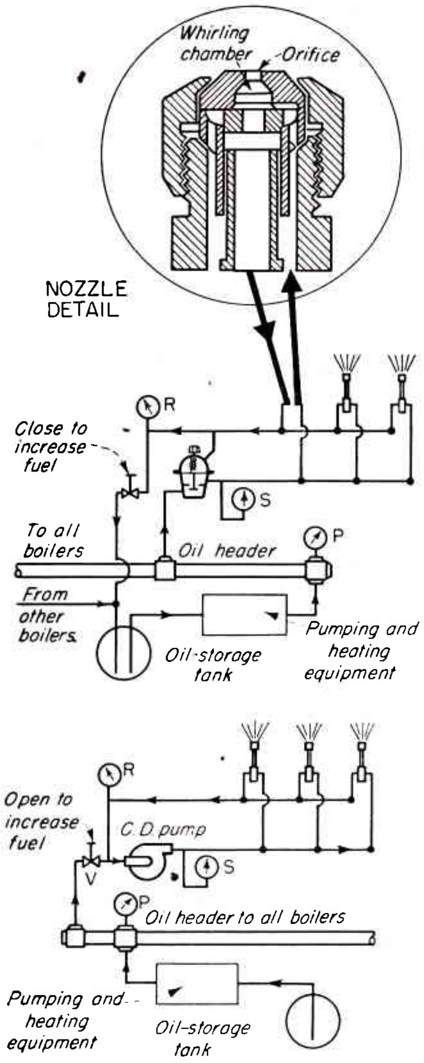


FIG. 6-16. Wide-range burner tip, above, can be regulated by either constant-differential valve or constant-differential pump to hold best atomizing pressure across supply and return.

6-12. Gas-burning Systems. Natural gas is the easiest of all fuels to handle. Extensive pipe-transmission systems carry the gas from the oil and gas fields to the consumer. For long-distance lines spanning up to 2,000 miles the gas is compressed to about 1,000 psig, but for local distribution systems this pressure is throttled to 200 psig or less. At the power plant gas tapped from the main flows through a gas-metering

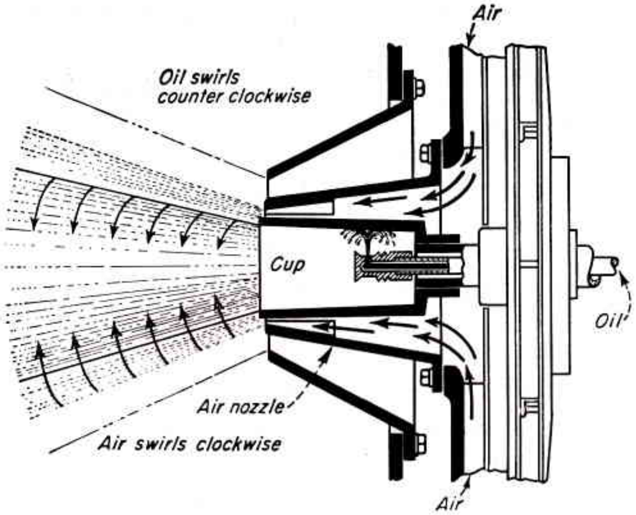


FIG. 6-17. Rotary-cup oil burner sprays oil on fast-turning cup to break up oil film at rim by centrifugal force. It mixes with secondary air spinning in opposite direction.

house. The branch supply line entering the house terminates in a header. Two parallel metering lines tap the header, each line consisting of stop valves, regulators, and meters. They join to a single line leading to the boiler room.

In the boiler room branch lines with metering orifices and pressure-regulating valves run to the individual burners in the boilers. Several burners serve each boiler to mix the gas with the proper amount of air. Figure 6-18 shows a center-diffusion tube design of gas ring using gas pressures of 2 to 25 psig. This is a combination oil and gas burner. It has an annular manifold between the air register and the furnace wall surrounding the burner opening. Orifices in the ring spray gas angularly across the incoming air stream controlled by the register. The mixed air and gas then enter the furnace to be heated to ignition by the flame and then burn.

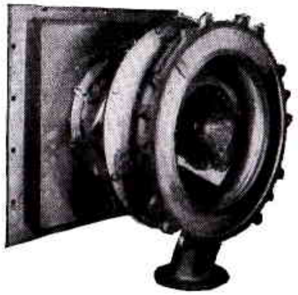


FIG. 6-18. Combination oil and gas burner receives natural gas through ring at right. Gas jets into air stream flowing through ring on way into furnace at right.

Only about 7 per cent excess air is needed to burn natural gas since it is in atomized form and has the maximum surface exposed for reaction with oxygen. The high hydrogen content of natural gas causes a high moisture loss to the vapor formed during combustion. This causes a lower boiler efficiency compared with other fuels with lower hydrogen content. But natural gas is the preferred fuel wherever it can be obtained. Most plants buy gas on a "dump" basis. When the gas-supply system must take all the available gas to feed higher-paying customers (usually domestic users), the gas is cut off on short notice to the power plant. The plant must then use an alternate fuel, usually oil, to meet its needs until the gas again becomes available.

PROBLEMS

6-1. Determine the energy used per day in kilowatthours to drive the pulverizer and fan feeding coal to a steam-generating unit based on the following data:

Pulverizer and fan performance:

Table P-1

Power input, kw	*Coal output, lb per hr*
28.0	4,000
38.1	7,000
48.6	10,000
63.0	14,000

Steam-generator fuel-rate data:

Table P-2

Steam output, lb per hr	*Fuel rate, lb per hr*
160,000	13,910
140,000	11,680
120,000	10,160
100,000	8,690
80,000	7,410
60,000	6,000

Steam-generator load-curve data (consider as straight lines between indicated points):

Table P-3

Time	*Output, lb per hr*
12:00 P.M.	50,000
7:00 A.M.	50,000
9:00 A.M.	140,000
5:00 P.M.	140,000
12:00 P.M.	30,000

6-2. A circular oil tank 30 ft long with a 4.5-ft diameter is used for oil storage. Calculate the number of days' oil supply the tank can hold for continuous operation at the following conditions:

Steam flow, 2,100 lb per hr
Steam, dry saturated at 200 psia
Feedwater temperature, 230 F
Boiler efficiency, 75.0 per cent
Fuel oil, 34°API

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CHAPTER 7

STEAM GENERATORS

7-1. Steam Generation. *Steam generators,* also called *boilers,* perform one of the major processes in a thermal-energy cycle: vaporizing water to steam. Steam may generate mechanical power or supply heat to a manufacturing process. The complex equipment in a steam generator evolved from the simple cauldron of Hero's engine of ancient history.

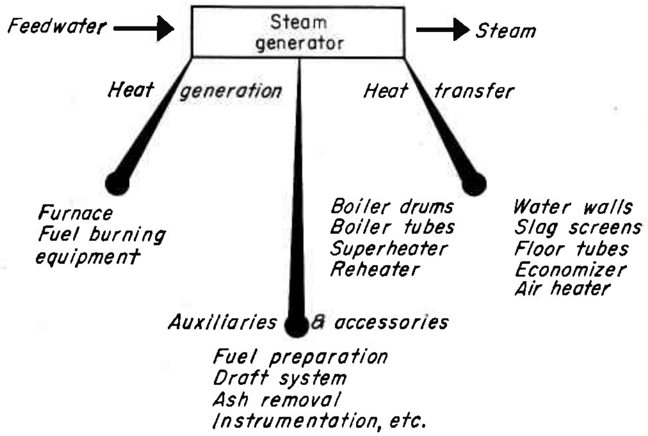


Fig. 7-1. Graphic representation of a steam generator. All units have furnace and fuel-burning equipment. Some or all of the components for heat transfer are used in units of various designs.

Figure 7-1 outlines the components and essential functions of a steam generator; the latter center on *heat generation* and *heat transfer*. Chapters 4 to 6 discussed heat generation with solid, liquid, and gas fuels in the furnace of a steam generator. Chapters 7 and 8 discuss the heat-transfer components, auxiliaries, settings, ratings, performance, and operation of the steam generator.

Thermodynamic and economic principles largely govern steam-generator design. Totally complete combustion and heat transfer cannot be justified in practice. Theoretically, combustion gases may be cooled to temperatures of incoming feedwater, steam saturation temperature, or incoming air depending on the heat-transfer components used. Economics controls the extent to which the design may approach theoretical limits.

For best cycle efficiency we should use the highest source temperature practically attainable. This controls partly the steam pressure and temperature selected.

Heat transfer in a steam generator decreases the availability of the energy released in the furnace. Energy released at combustion temperatures up to 2800 F transfers to water at less than 705 F and steam at less than 1100 F. The metallurgical limits of steel prevent using working temperatures much above 1100 F; hence we must accept the large loss in availability in present-day steam generators.

Steam generators transfer heat by radiation, conduction, and forced convection. Some components "see" the furnace flame, to receive heat mostly by radiation but also by conduction and forced convection. Other components depend on only conduction and convection.

Efficiencies of steam generators vary widely with basic design and operating conditions. In the large units efficiencies higher than 90 per cent have been reported.

The ASME Boiler Construction Code specifies the limits needed for safe construction. The Code, originated in 1914, set up standards that must be met and formed the basis for legal requirements. The Code has several sections—Power Boilers; Material Specifications; Boilers of Locomotives; Low-pressure Heating Boilers; Unfired Pressure Vessels; Welding Qualifications; Suggested Rules for Care of Power Boilers. Periodic revisions keep the Code in step with modern needs—addenda sheets and interpretations are published as new situations arise. Most states accept the Code as law or as a basis for state boiler laws. The ASME Test Code for Stationary Steam Generating Units outlines the precise methods to be used in making performance tests and determining heat-transfer surfaces.

7-2. Circulation Principles. The water within and the feedwater entering a steam generator circulate through given tubes in most units. This water vaporizes to steam bubbles, to leave as wet, dry, or superheated steam for use in an engine or a heating process.

Natural Circulation. Water circulates naturally in a boiler when its density in one part of a circuit is less than in another part at the same level. The boiler drums, tubes, and waterwalls make up the multipassaged circuit. Figure 7-2 shows the natural circulation in simple drum and water-tube circuits. In Fig. 7-2*a* water near the drum interior surface becomes heated and forms steam bubbles, lowering its density. The water in the drum center, being heavier, displaces the lighter water-bubble mix and in turn becomes heated. This sets up a constant circulation that releases steam into the upper drum region.

In Fig. 7-2*b* the water from the drum flows down the *downcomer* tube to the bottom of the heated *riser*. Heat forms steam bubbles in the water passing through the riser, lowering the density of the mixture. The heavier downcomer water displaces the riser mixture to establish continuous circulation as long as heat enters the riser.

The per cent of steam by weight at the riser top is called the *top dryness;* the reciprocal of this term is the *circulation ratio.* Natural circulation largely depends on the height of riser and downcomer and the mean density of the fluids in them. These depend on the steam pressure, dryness in the tubes, friction losses in tubes, entrance, exit, bends, and valves, and fluid acceleration. Fluid velocities in the tubes depend on the balance between losses and accelerations on the one hand and the available pressure difference in riser and downcomer on the other hand.

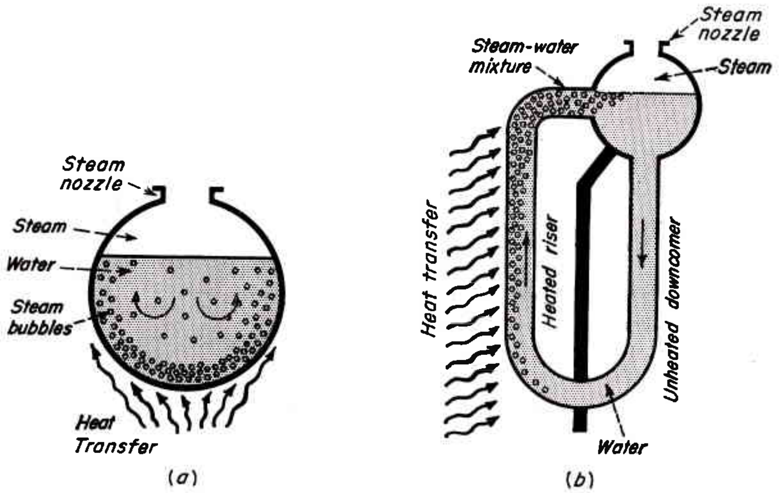


FIG. 7-2. (*a*) Steam formation in simple drum. Steam bubbles formed at heated surface rise and are released to steam space. Heavier water displaces bubbles. (*b*) Simple water-tube circuit. Note steam-water mixture in heated *riser* of lesser density than water in unheated *downcomer*. Bubbles are released to steam space.

Figure 7-3 shows that increasing boiler pressure lowers the pressure difference of riser and downcomer. In Fig. 7-2 the steam breaks out of the water in the drum to accumulate in the vapor space and leave through the steam nozzle. Entering feedwater equals the steam leaving and aids natural circulation. The water level in the drum remains fairly constant when steam and feedwater flows are equal.

Figure 7-4 shows the natural-circulation principle in a typical *straight-tube* and *bent-tube* boiler. Natural circulation has been largely used in boilers up to about 2,600 psi pressure and ordinarily produces enough velocity for effective heat transfer.

Forced Circulation. Air heaters, economizers, superheaters, and reheaters normally use forced circulation. In boilers with steam drums a pump taking suction at the bottom of the downcomer and discharging into the riser produces forced circulation. The assured velocity aids heat transfer and avoids burning out the tubes from overheating.

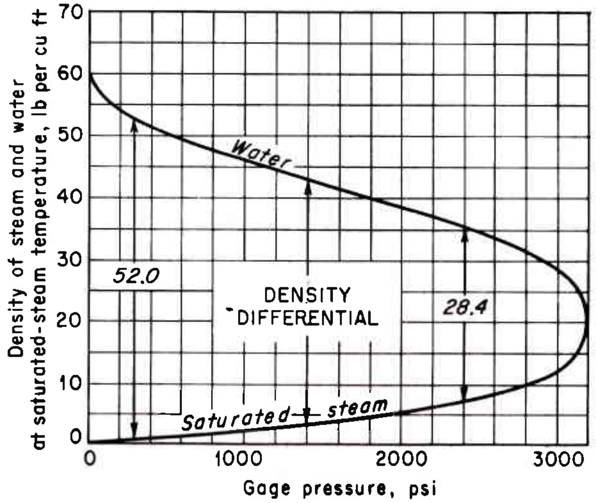
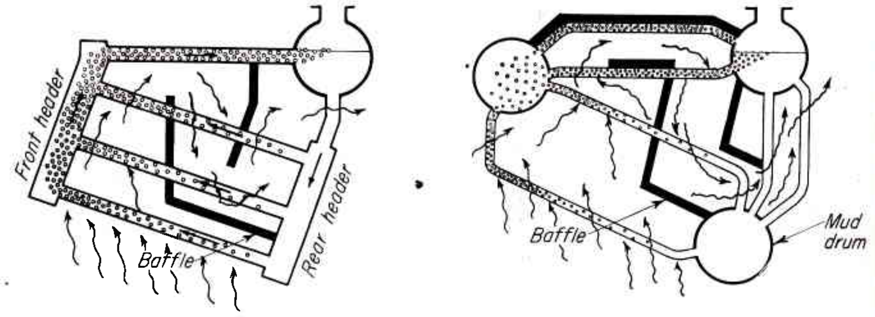


FIG. 7-3. Variation of saturated water and steam density with pressure. Note density in riser is between lines; exact value depends on average dryness in riser.



(*a*) (*b*)

FIG. 7-4. (*a*) Straight-tube boiler circuit. Basic idea of natural circulation applies, but downcomer tubes and rear header are not heated enough to cause steam formation. (*b*) Bent-tube boiler circuit. Circulation is more complex in multidrum boiler. Rear tubes act as downcomer to mud drum. Steam bubbles form in inclined tube banks.

In drumless steam generators the feedwater pump forces the water in at one end of a series of parallel tubes, exposed to the furnace and combustion gases, to emerge as superheated steam at the other end. This type of steam generator must be used near or above the critical point (Fig. 7-3). Controlled forced circulation gives latitude in arranging tubes and drums. Orifices at the inlet end of paralleled riser tubes control flow distribution.

Forced-circulation boilers use thin-walled tubes of smaller diameter, $1\frac{1}{4}$ to $1\frac{5}{8}$ in. These units reach operating conditions in a shorter time, they have about half the weight of steel, and a ruptured tube does not constitute so great a hazard. Pumping energy, sludging of small tubes, and investment in extra equipment may be a disadvantage. European plants have used forced circulation for many years—controlled-circulation units are now finding favor in the United States.

The steam-water circulating rate must be proportioned to the heat transfer and gas flow to prevent overheating tube metal or to avoid only partial vaporization. Too high a dryness ratio may overheat the tubes. Figure 4-10 shows that steam films have a higher resistance than water films, and so the intermediate metal temperature may rise too high and produce creep stressing and burnouts.

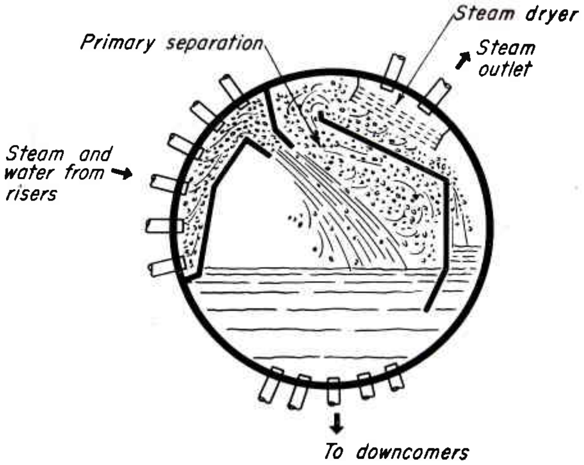


FIG. 7-5. Simple baffling used for separation and drying.

Designing boiler circulation depends on experience and empirical concepts combined with theory. Circulation ratios vary from about 6 to 25 lb water per pound steam; tube velocities vary from about 1 to 7 ft per sec, depending on tube inclination; vertical tubes use lower values.

7-3. Steam Separation and Purification in Drums. Risers discharge a mixture of steam, water, foam, and some sludge to boiler drums. The steam must separate from the mixture before leaving the drum. Moisture in steam reduces superheat, corrodes the superheater, and damages the steam turbine. The moisture droplets (*carry-over*) in steam contain dissolved salts from the boiler water. When the droplets vaporize, the salts deposit out in the superheater, turbine, valves, and traps. The droplets decrease heat transfer, turbine efficiency, and control.

At pressures less than 300 psia the large difference in water and steam densities aids separation by gravity, provided that there is enough water-surface area for steam-bubble release. At elevated pressures mechanical

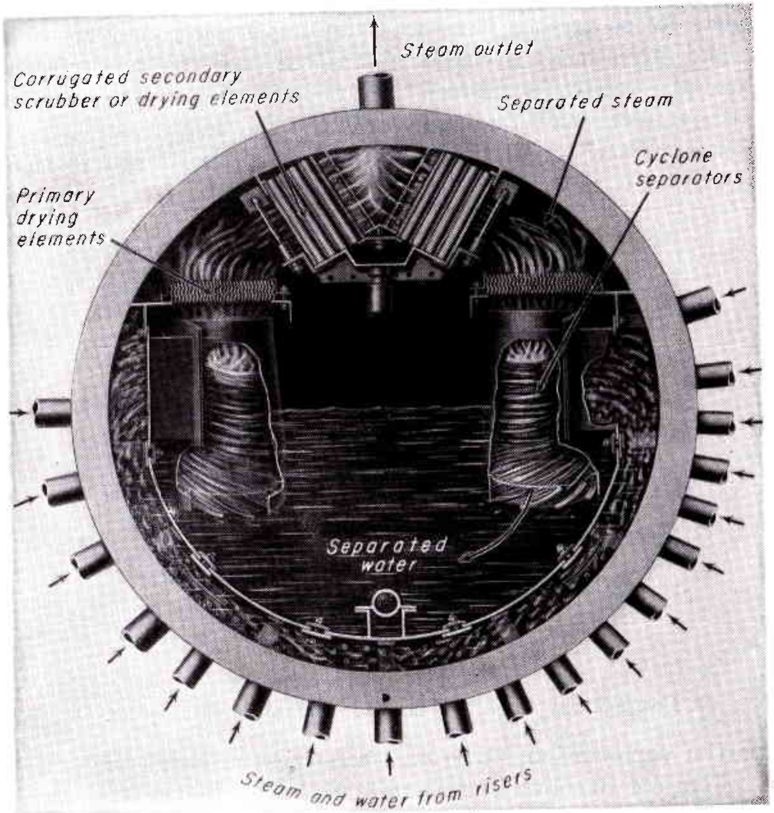


FIG. 7-6. Cyclone separators in boiler drum, stationary cylindrical cyclones employ centrifugal force to effect primary separation of steam-water mixture from risers. Primary drying elements precede secondary scrubbers in steam-flow path leaving drum. Single or double rows of cyclones may be used. *(Courtesy of The Babcock & Wilcox Company.)*

separators in the drum must be used, such as baffles or cyclones. These are called *drum internals;* they control these sequential processes: (1) primary separation of steam from water, (2) steam washing, and (3) steam drying.

Primary separation removes the major part of the circulating water, foam, and sludge from the steam. In washing, the steam bubbles through incoming feedwater or passes through a spray of feedwater

(low solid content). This dilutes the moisture droplets still carried by the steam. Dryers mechanically separate small droplets from large masses of flowing steam.

Primary separation must be used in high-pressure boilers. Washing and steam drying will be used to attain high purification when needed. Figures 7-5 and 7-6 show some types of separators and dryers. The drum internals stand in the drum above normal water level. Single-drum

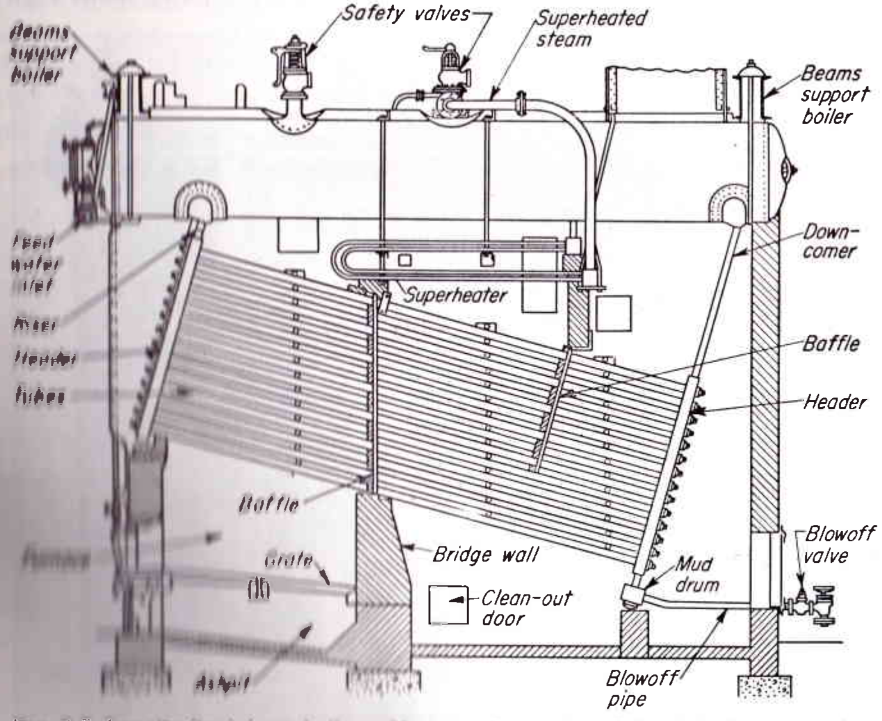


FIG. 7-7. Longitudinal-drum boiler. Note gas-flow path and simple boiler accessories.

boilers often use only separation and drying. The steam may be separated in one drum and washed and dried in another drum in multidrum boilers. These sometimes use a *dry drum* above the water-holding main drums. Steam from the main drums flows through it for drying on the way to the superheater; separated water drains to the main drum.

A boiler is priming when it has large amounts of carry-over, often caused by too high water level. Poor water quality may cause foaming and an increase in carry-over. Foaming reduces the effectiveness of drum internals; this seldom occurs with modern feedwater conditioning.

Continuous or periodic blowdown removes boiler water from the unit and lowers the boiler-water impurities. Surface blowdown removes scum from the water surface. Bottom blowdown removes sludge and dissolved solids.

7-4. Water-tube Steam Generators. These units are universally used in the larger capacities. Water flows inside tubes and drums and receives heat by radiation and conduction from the furnace and combustion gases flowing over the tube surfaces. Positive fluid circulation assures good heat transfer and enables building large-capacity units. Tubes and drums smaller than in fire-tube units (Sec. 7-5) make high steam pressures feasible. Tubes, external to the drums, can be placed for best heat absorption. Scale and deposits can be removed more easily from inside tubes than from the outside. Water-tube boilers may be classified in several ways: (1) straight or bent tube; (2) longitudinal or cross drum; (3) sectional or box header; (4) one or more drums; (5) cross or parallel baffles; and (6) horizontal, vertical, or inclined tubes.

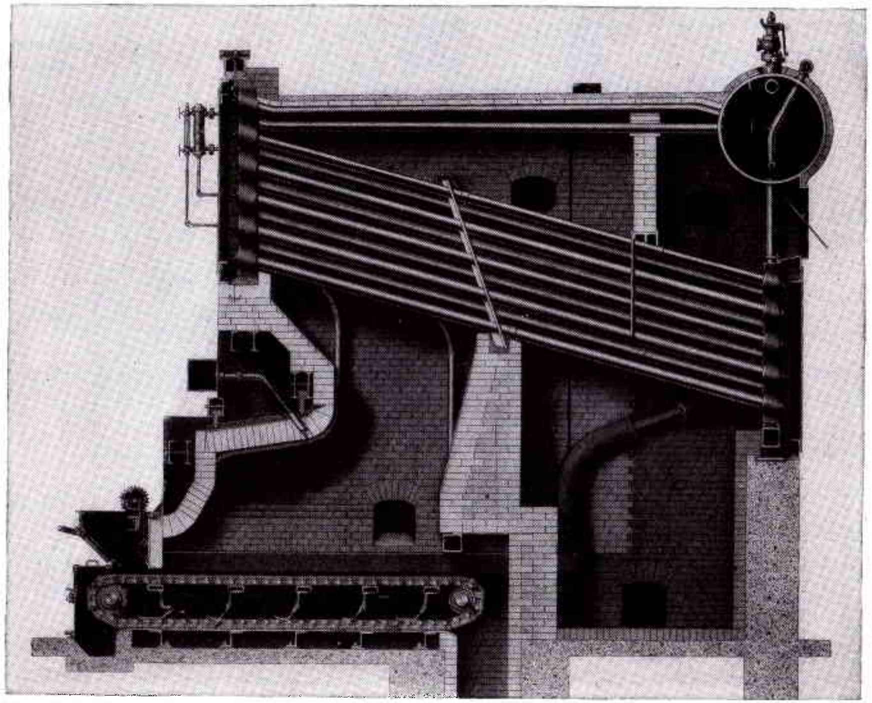


FIG. 7-8. Cross-drum boiler. Note relatively lower headroom and baffling to create a *three-pass* boiler. (*Courtesy of The Babcock & Wilcox Company.*)

Straight-tube Units. Figure 7-7 shows a longitudinal-drum boiler with a simple hand-fired grate and superheater tubes. Headers connect to the drum bottom at each end. In simplest form these headers consist of narrow steel boxes. Inclined tubes in parallel connect the two headers. The water level stands normally at about drum center. Combustion gas flows over the tubes in several passes formed by baffles and discharges to the stack.

Denser drum water flows down the downcomer to displace the lighter water in the tubes partly evaporating to steam bubbles. A mixture of water and steam bubbles flows up through the risers into the drum, where the steam separates from the water. Older boilers have the downcomer and riser as box headers that must be stayed internally to prevent rupture.

Figure 7-8 shows a cross-drum boiler with *sectional* headers. The drum axis stands at right angles to the tube axes. Each of the sectional headers, made of individual steel seamless box headers, carries one or two vertical rows of tubes. The sinuous shape of some headers staggers tube positions in the vertical direction. Handholes in the headers opposite each tube give access for tube cleaning and replacement.

The bubble-laden water from the tubes flows through the cross tubes to discharge into the drum. This boiler has a traveling-grate stoker from which the combustion gas flows through the three passes.

Figure 7-9 shows one of the earlier (1925) modifications of the simple cross-drum boiler for pulverized-coal firing. Such boilers run at pressures up to 1,400 psig.

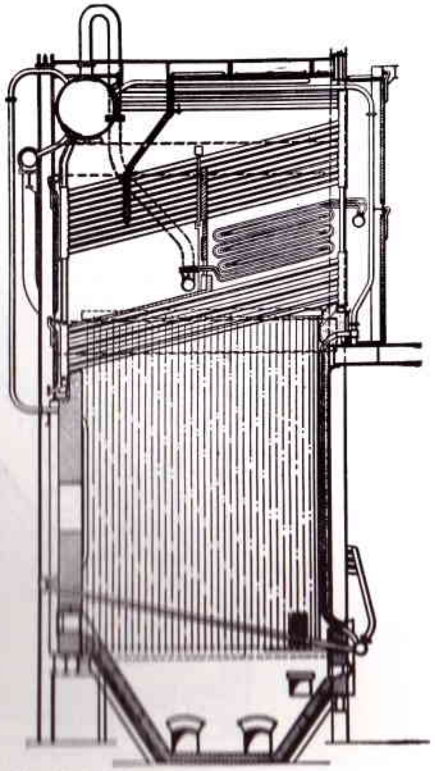


FIG. 7-9. Adaptation of cross-drum sectional-header design to higher-capacity units. Design data are: capacity 175,000 lb per hr, pressure 488 psi, steam temperature 750 F, and boiler surface 11,480 sq ft. (*Courtesy of Combustion Engineering Company, Inc.*)

Bent-tube Units. These units have several drums connected by bent tubes. Eliminating headers reduces the maintenance costs and the handling of large numbers of handhole caps and gaskets. Tubes are cleaned from inside the drums, and only one or two manholes need be removed to get inside. Bent tubes can be arranged in a variety of ways to assure good circulation (Fig. 7-10). Drier steam can be obtained by taking it from the rear drum. This type also has water-cooled walls.

Figure 7-11 shows a low-headroom boiler; water circulates from the uppermost drum down to the lower drum, up to the intermediate water drum, and across to the upper drum. Baffles direct the gas flow through three passes.

Early water-cooled-wall tubes were independent of the main boiler tube circuit. Modern design treats all tubes as a unit; Fig. 7-12 shows one form called an *integral-furnace* steam generator.

Waterwalls fitted to Stirling boilers allow higher combustion rates and increased capacity. Waterwalls replace refractory walls, which limited firing rates. Figure 7-13 shows a bent-tube boiler with waterwalls, an extension of the unit in Fig. 7-10. Feedwater enters the rear drum of the three upper steam-and-water drums. Denser drum water circulates down the rear tube bank to the low mud drum and then up both the center and front tube banks to the upper drums. Baffling leads the combustion gas into intimate contact with the tubes.

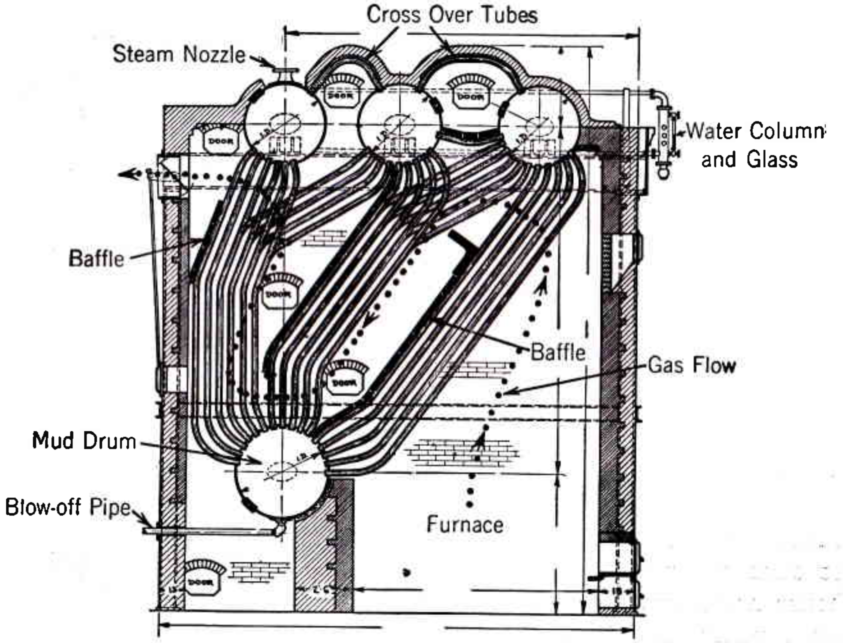


FIG. 7-10. Simple multidrum bent-tube boiler. Commonly called a Stirling boiler. (*Courtesy of Murray Iron Works Company.*)

Steam generators of this type for high-pressure and -temperature service must meet specific conditions and are "tailor-made." While details vary from unit to unit, the boiler convection surface is a small percentage of the total, which includes waterwall, superheater, air heater, and economizer surfaces. Some radiant designs have no convection surface; the waterwall surface allows high heat-transfer rates in the radiant zones. As steam temperatures increase, the superheater has an increasing percentage of the total surface. Air heaters and economizers cool combustion gases to develop over-all high efficiency.

Natural-circulation units must have considerable height to ensure positive steam-water flow. Suspending these units from steelwork allows them to expand downward as they heat up. Figures 7-14 to 7-16 show a variety of modern large steam generators. Steam pressures range from 900 to 2400 psig and temperatures from 800 to 1100 F.

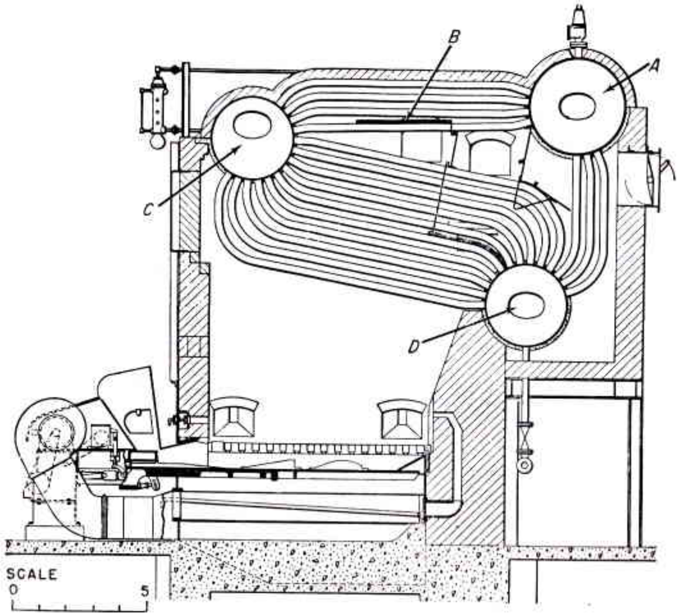


FIG. 7-11. Three-drum bent-tube unit for low headroom. Short rear tubes connect steam drum and mud drum. *A*, steam and water drum; *B*, baffle; *C*, water drum; *D*, mud drum. (*Courtesy of Combustion Engineering Company, Inc.*)

7-5. Fire-tube Boilers. Most small boilers use the type of construction in which the combustion gases pass through interiors of tubes submerged in the water to be heated. The furnace may be separate or built into the boiler shell (see Fig. 7-2*a*).

Fire-tube boilers may be classed as: (1) externally or internally fired, (2) through or return-tube type, and (3) horizontal, vertical or inclined.

Figure 7-17 shows a horizontal-return tubular (HRT) fire-tube boiler, built for steam pressures up to 250 psig with a maximum of about 4,000 sq ft heating surface. The boiler has a horizontal drum on top of a furnace surrounded by a refractory setting. Gases rising from the fuel bed on the grates flows under the drum and around the protecting pier to enter the fire tubes piercing the drum. The gases give up most of their heat in flowing to the smokebox at the boiler front.

The drum holds enough water to cover all the tube exteriors. Heating the water forms steam bubbles on the tube surfaces that rise and induce a circulating flow inside the drum over the tubes. Feedwater enters the drum to replace the steam leaving at the top, where the steam collects as the bubbles burst through the water surface.

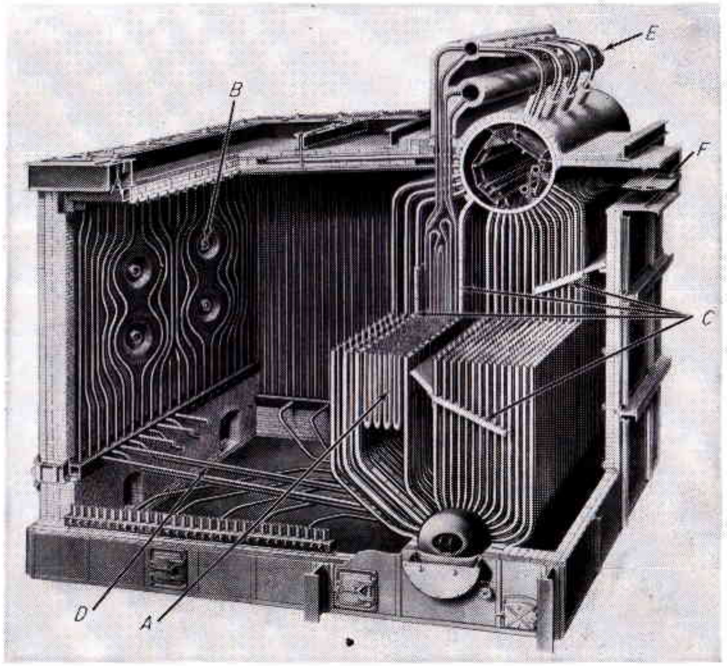


FIG. 7-12. Standardized steam-generating unit of integral-furnace type. Capacities of 30,000 to 350,000 lb per hr; pressures to 1,000 psi; suitable for pulverized-coal, stoker, oil, or gas firing. *A*, superheater; *B*, burners; *C*, baffles; *D*, floor tubes; *E*, superheated steam header; *F*, damper. (*Courtesy of Combustion Engineering Company, Inc.*)

Doors in the front of the smokebox give access to the tubes for cleaning. Either structural steel or the refractory furnace setting supports the drum. Scale deposits on the water side of the tubes cannot be easily cleaned off.

Other designs of fire-tube boilers include: (1) *simple vertical boilers,* (2) *two-pass boilers,* (3) *Scotch marine boilers,* and (4) *locomotive type boilers.* The simple vertical boiler is an HRT set on end with the furnace underneath. A two-pass boiler has the gases first flowing through a lower set of tubes in the drum and then returning for a second pass through a higher set of tubes. Scotch marine boilers have a metal-walled furnace built into the drum along with the fire tubes. Locomotive boilers have a firebox forming one end of the drum.

Modern fire-tube boilers include the *packaged boiler* built for pressures up to 250 psig and capacities up to 600 hp. These self-contained units have fans, feed pump, furnace, drum, and automatic controls all mounted on a single base (Fig. 7-18). The furnace resembles that of a Scotch marine boiler but has three or four gas passes through the drum. The mechanical-draft fans overcome the friction losses induced by the long multipass gas circuit. The hinged back can be swung open to work on the rear tube sheet.

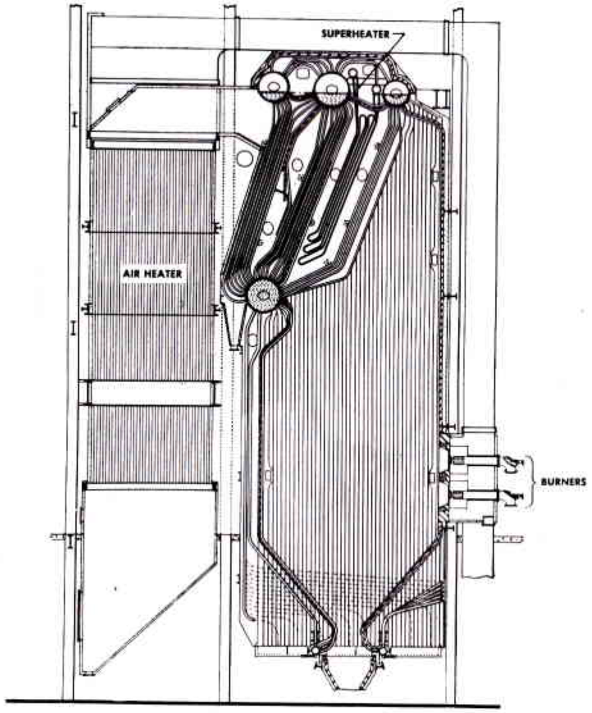


FIG. 7-13. Water-tube walls adapted to Stirling boiler. Capacity 250,000 lb per hr, 650 psia, 750-F superheater outlet. (*Courtesy of The Babcock & Wilcox Company.*)

7-6. Steam-generator Accessories. These units need equipment for adequate control and operation, such as:

Water-level Gauges. These indicate the water level in drums. Low-pressure boilers use simple tubular glasses (Fig. 7-8). Higher pressures need more sturdy construction, which uses flat glass plates in metal housings. For remote observation optical systems with periscopic mirrors, electric transmission, and television methods may be used.

Safety Valves. These valves are loaded by springs and other methods to automatically relieve overpressures in the drum, superheater, and reheater. Superheater relief valves are set to open before drum relief valves to keep steam flowing through the superheater and so prevent tube burnouts.

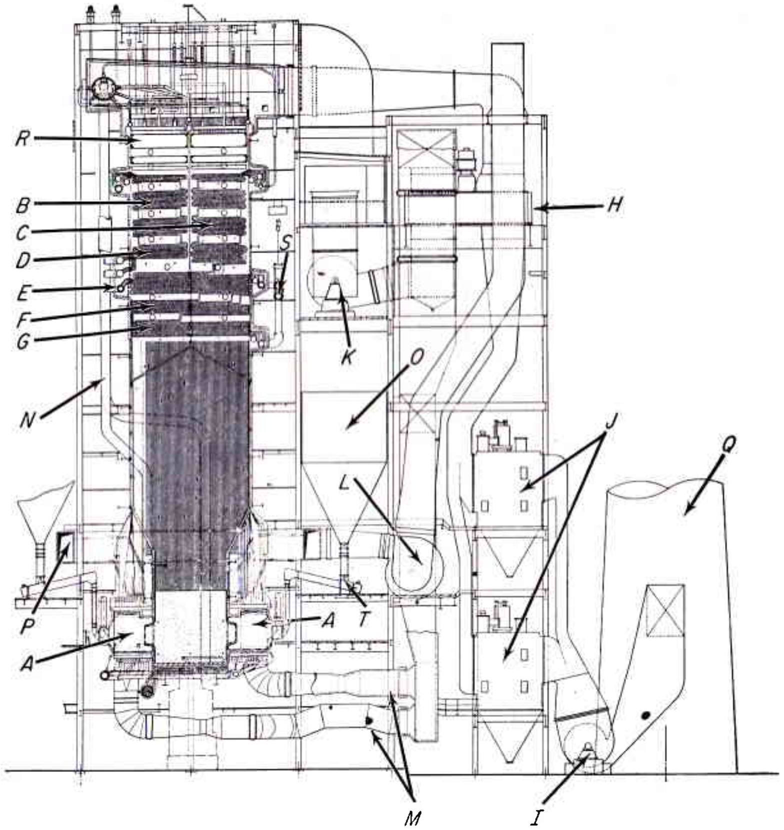


FIG. 7-14. Typical cyclone-furnace-fired radiant reheat boiler for a central station. Capacity 2,100,000 lb per hr, design pressure 2,700 psi, superheater outlet pressure 2,450 psi, high-pressure steam temperature 1050 F, reheat steam temperature 1000 F. *A*, cyclone furnace; *B*, reheat superheater; *C*, primary superheater; *D*, reheat superheater; *E*, primary superheater; *F*, secondary superheater; *G*, secondary superheater; *H*, air heater; *I*, induced-draft fan; *J*, precipitators; *K*, forced-draft fan; *L*, gas-recirculating fan; *M*, secondary-air ducts; *N*, downcomers; *O*, coal bunker; *P*, gas-recirculating duct; *Q*, stack; *R*, economizer; *S*, attemperator piping; *T*, feeder. (*Courtesy of The Babcock & Wilcox Company.*)

Soot Blowers. Pipe and nozzle systems blow jets of air or steam over the fire sides of heat-transfer surfaces to remove soot and ash deposits. Retractable systems that can completely withdraw from the boiler passes and furnace prove to need less maintenance.

Feedwater Regulators. These controls hold drum water level between safe limits during constant and varying loads. They must compensate for the change in total boiler water with load; a higher percentage of steam bubbles at high steaming rates reduces the amount of water needed in the boiler and tends to raise the drum water level.

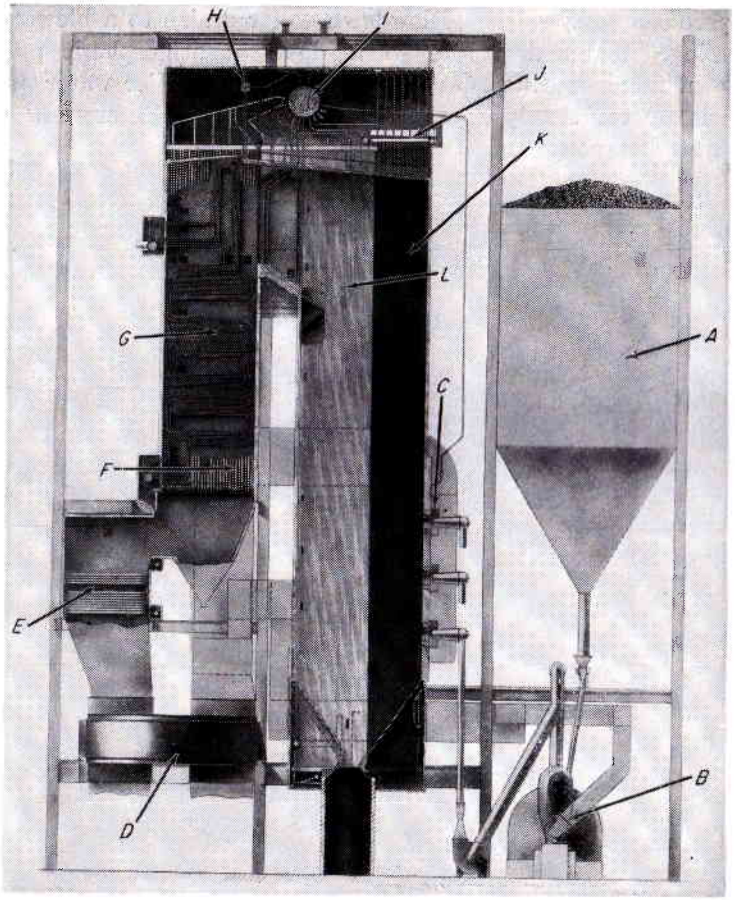


FIG. 7-15. Typical separated furnace, natural-circulation differential fire-controlled reheat steam generator for a central station. Above is the reheat furnace, and the companion furnace (not shown) is the superheat furnace in which the convection superheater replaces the reheater *G*. Capacity 1,850,000 lb per hr, pressure 2,150 psi at superheater outlet, 1010 F primary steam temperature, 1010 F reheat temperature. *A*, coal bunker; *B*, ball-mill pulverizer; *C*, burners; *D*, regenerative air heater; *E*, economizer; *F*, waterwall surface; *G*, reheater; *H*, superheat-control condenser; *I*, steam drum and internals; *J*, superheat-control condenser; *K*, radiant superheater; *L*, waterwalls. (*Courtesy of Foster Wheeler Corporation.*)

Blowdown Valves. Valves in the lines connected to the lowest part of the boiler water circuit drains off water containing solids under full boiler pressure. Lines placed at water level skim off any foam, controlled by *surface blowdown valves.* Blowdown water drains to a blowoff tank for cooling. Water drains to waste, later at atmospheric pressure. Blowdown removes salts that accumulate in the boiler water and are carried in by the steady flow of feedwater. The salts are left behind as the water evaporates.

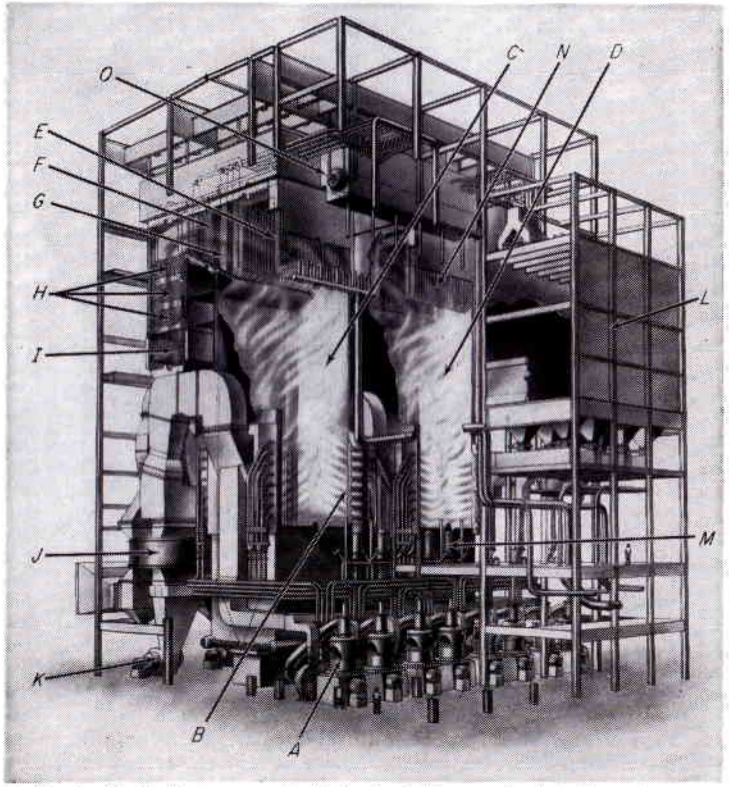


FIG. 7-16. Typical twin-furnace controlled-circulation central-station steam generator. Capacity 1,800,000 lb per hr, pressure 2,100 psi, temperature 1050 F, reheat temperature 1050 F, height 160 ft, twin-furnace volume 149,000 cu ft, fuel consumption 2,940 tons per day. *A*, pulverizers; *B*, tilting tangential burners (one per corner with eight coal nozzles each); *C*, furnace (reheat side); *D*, furnace (superheat side); *E*, reheater, low-temperature stage (channel); *F*, reheater, intermediate stage (pendent); *G*, reheater, high-temperature stage (pendent); *H*, superheater, low-temperature stage (three banks); *I*, economizer; *J*, Ljungstrom air preheater; *K*, f-d fan; *L*, coal bunker; *M*, circulating pumps; *N*, superheater, low-temperature stage (channel); *O*, steam drum. (*Courtesy of Combustion Engineering Company, Inc.*)

Television. Closed-circuit television systems allow remote viewing of water-level gauges, smoke discharges from stacks, and flame conditions in furnaces. Viewing screens are mounted in central control rooms.

7-7. Advanced Designs. The modern need for high efficiency and the ability to burn low-grade fuels, reducing the amount of high-cost mate-

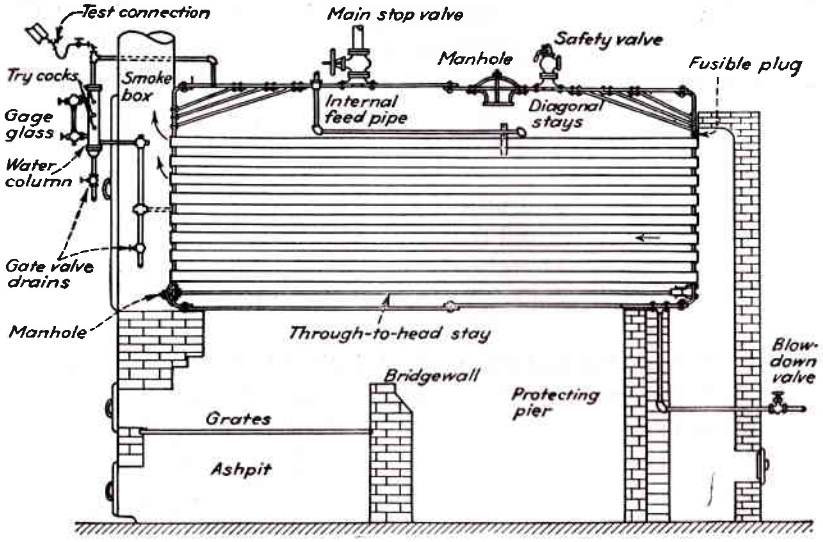


FIG. 7-17. Horizontal-return tubular boilers (HRT) are commonest of fire-tube types. Gas flows along the bottom of the shell to the rear and then forward through the fire tubes to a smokebox at the front.

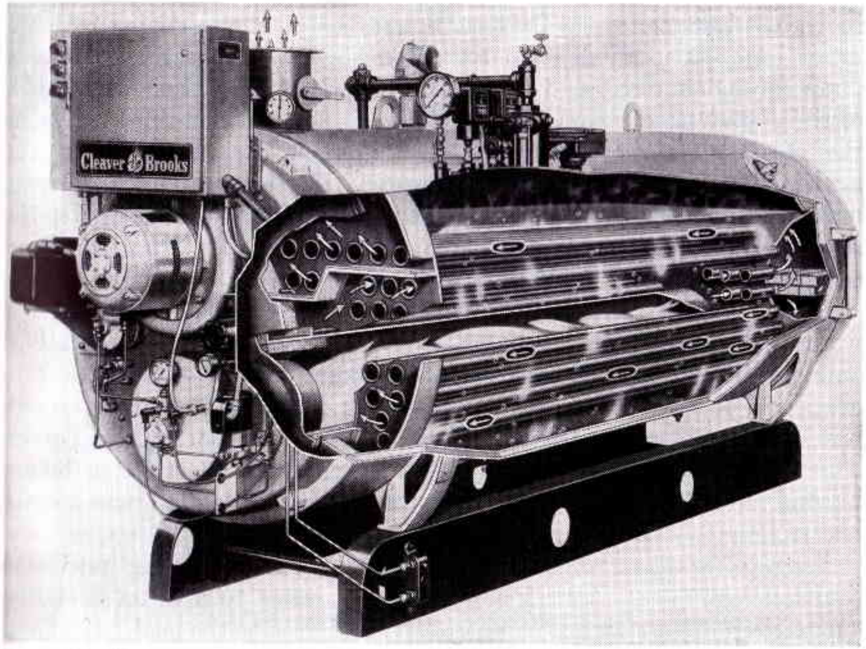


FIG. 7-18. Typical modern self-contained fire-tube boiler. Note the four-pass construction (one through furnace, three through tubes) and accessibility through front and rear hinged doors. Designed for oil or oil-gas combination firing. (*Courtesy of Cleaver-Brooks Company.*)

rial, has led to designing steam generators here and abroad that differ from past conventional practice. Features that make these units different include:

- Forced circulation
- Small tubes
- Lack of drums
- Pressurized combustion
- Multiple-tube circuits
- Extreme compactness and low weight

Figure 7-19 shows the features of the principal boiler types.

La Mont Boiler. This forced-circulation unit uses small-diameter tubes in the evaporating section. This gives flexibility in placing the heat-transfer surface for best advantage. The drum may be placed well away from the furnace to ease the design problem. The small tubes give a unit with high specific output which can be used for high or low pressures. This type of unit finds increasing acceptance in the United States as the *controlled-circulation* boiler.

Loeffler Boiler. Feedwater passes through an economizer on its way to a drum external to the furnace. Superheated steam mixing with the drum water evaporates it to saturated steam. The saturated steam flows through radiant and convection superheaters. About two-thirds of the superheated steam returns to the drum, and one-third leaves as the steam-generator output. A steam-circulating pump forces the saturated steam through the superheater. Pressures about 2,000 psig make this boiler economical, and only small-bore tubes need be used in the superheater radiant section. These units can handle drum salt concentrations as high as 8,000 ppm without trouble.

Benson Boiler. This drumless once-through boiler takes boiler feedwater in at one end and discharges it as superheated steam at the other. In Fig. 7-19 feedwater flows through the radiant parallel-tube section T to evaporate partly. The steam-bubble-water mixture then flows through the lower-temperature transition section TS to evaporate completely. The steam flows through the higher-temperature superheater before leaving the unit. This lightweight boiler can be started in about 15 min and is designed for pressures of 400 to 3,500 psig.

Schmidt-Hartmann Boiler. This multiple-circuit boiler mitigates feedwater troubles. A feed pump supplies water to a drum that discharges saturated steam to a superheater and then to the load. A closed secondary circuit heated by the furnace resembles a sectional header boiler. This section supplies the heated vapor in a coil to evaporate the main feedwater. The boiler performs well in producing dry steam

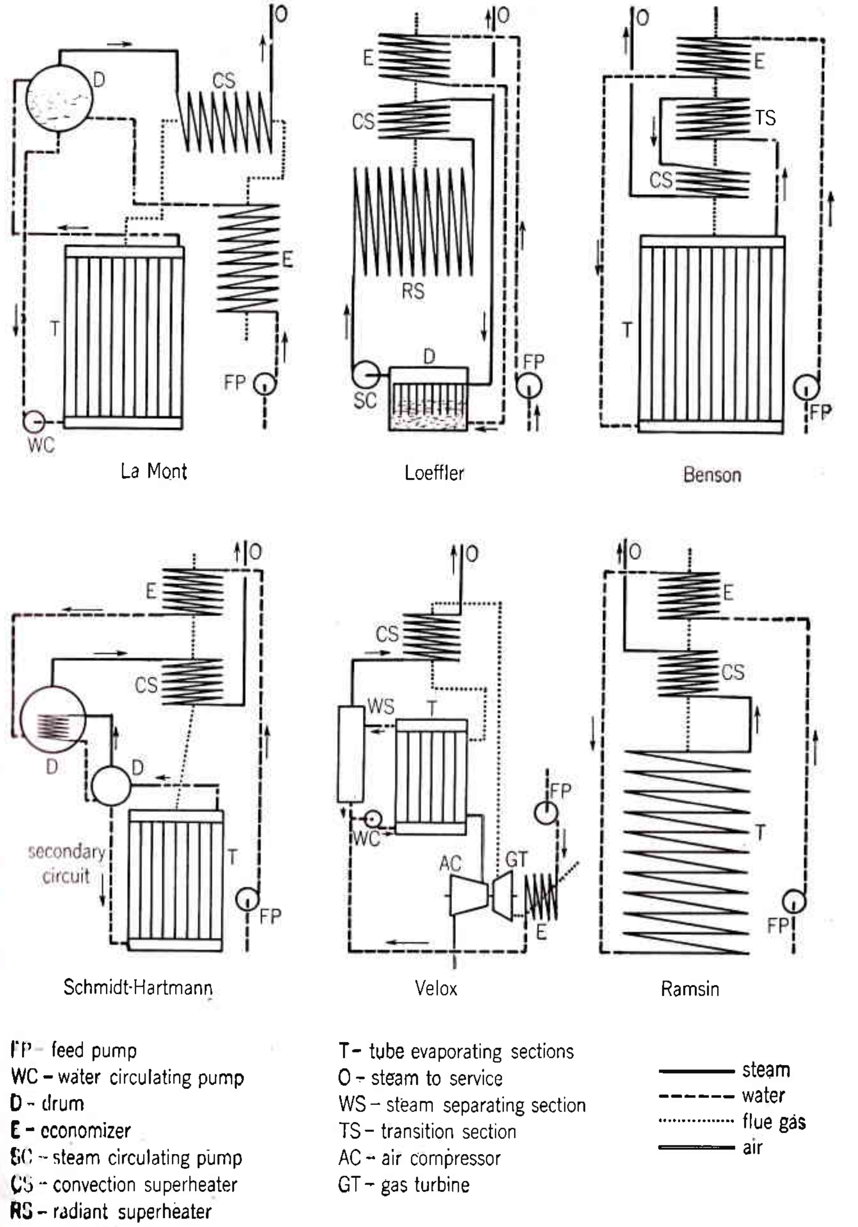


FIG. 7-19. Diagrammatic representation of various special boiler designs. *La Mont* and *Benson* principles are finding present-day use in controlled-circulation and supercritical boilers.

which eliminates carry-over. The boilers are built for pressures ranging from 500 to 1,800 psig.

Velox Boiler. The pressurized combustion chamber (furnace) of this boiler uses low excess air and has high heat-transfer rates. The furnace runs at about 35 psig pressure; the combustion gases after heating the water and steam pass through a gas turbine to atmosphere. The gas turbine drives the axial-flow compressor that raises incoming combustion air from atmospheric to the furnace pressure. This compact unit develops high efficiency and is built for pressures ranging from 200 to 1,200 psig and 950 F.

Ramsin Boiler. This Russian once-through boiler has inclined coils T arranged in spiral form; 40 coils are paralleled around the furnace. Steam generated in the headers flows into headers and then to the superheater. The boilers work at various pressures and are built of standardized parts.

The supercritical-pressure boilers now being built in the United States use the once-through principle; these will be discussed in Chap. 20.

7-8. Furnace Design. Furnace design depends on the steam-generator operating conditions, heat-transfer principles, combustion process, available materials, and empirical, or experience, factors. Furnace volume and the proportions selected are fixed by: (1) the type of fuel; (2) steam capacity and conditions—maximum load, load range, pressures, and temperatures; (3) per cent of heat recovery that proves economical; (4) type of firing; (5) excess air; (6) flame length; (7) ash-fusion temperature; (8) stoker or grate area; and (9) furnace-wall and arch constructions.

The following relations fix the primary factors of the furnace:

$$w_F \times HHV \times \eta_g = \Sigma w_s \, \Delta h \tag{7-1}$$

where w_F = fuel firing rate, lb per hr
HHV = fuel heating value, Btu per lb
η_g = over-all steam-generator gross efficiency, fraction
$\Sigma w_s \, \Delta h$ = total heat transfer to cycle fluid in steam generator (boiler, superheater, economizer, reheater), Btu per hr

This equation must use an estimated efficiency that depends on the amount of heat recovery in the unit.

Furnace volume partly controls the furnace heat-release rate,

$$H_{Rv} = w_F \times \frac{HHV}{V} \tag{7-2}$$

where H_{Rv} = furnace heat-release rate, Btu per cu ft-hr
V = furnace volume, cu ft

The heat-release rates chosen by the designer depend on the fuels and previous experience with those fuels in existing furnaces of similar type (Table 7-1). The heat-release rates used also depend on whether the furnace has air- or water-cooled walls.

TABLE 7-1. FURNACE HEAT-RELEASE RATES

Fuel	Type of firing	H_{Rv}, Btu per cu ft-hr
Coal	Stationary grate	40,000–50,000
Coal	Stoker	15,000–50,000
Coal	Pulverized	15,000–45,000
Oil or gas	Burner	15,000–50,000

The heat release in terms of heat-transfer surface also controls furnace proportion,

$$H_{Ra} = w_F \times \frac{HHV}{A_F} \tag{7-3}$$

where H_{Ra} = heat-release rate, Btu per sq ft-hr
A_F = area of projected furnace envelope, sq ft

Typical values vary around 80,000 to 90,000 Btu per sq ft-hr for a furnace with radiant flames. This ratio is more significant than the volumetric heat release [Eq. (7-2)].

For stokers and grates:

$$w_F = A \times C_R \tag{7-4}$$

where A = projected grate or stoker area, sq ft
C_R = combustion rate, lb fuel per sq ft-hr

The value of the combustion rate varies with type and size of coal, furnace draft, and type of stoker or grate. This equation gives the plan area for stoker-fired furnaces; knowing the volume from Eq. (7-2), the designer can approximate the dimensions. These may be modified by circulation needs in the boiler and the placement of arches in the furnace to control combustion. In pulverized-coal-, oil-, and gas-fired furnaces the boiler circulation needs determine the furnace height, and the plan area can be found with the aid of Eq. (7-2). The type of burner may modify the furnace cross-sectional area.

7-9. Furnace Construction. Smaller boilers use solid refractory walls, but they will be air-cooled in larger units. Larger central-station type boilers have water-cooled furnaces that absorb up to 60 per cent of the heat transferred. The waterwalls of modern boilers may contain all their evaporating capacity.

Simple furnace walls have an interior face of refractory firebrick, an intermediate layer of insulation with the exterior *casing* made of sheet metal. Refractory may be in several forms: brick or cast or built-up monolithic covering. Refractory material contains about 40 per cent aluminum oxide (Al_2O_3), 55 per cent silica (SiO_2), and 5 per cent other oxides as impurities. Fusion temperatures of refractories vary from 2700 to 3600 F and limit the maximum fire temperature that may be used. Insulating materials as bricks or fill may be asbestos, 85 per cent magnesia, rock wool, diatomaceous earth, or any mixture of these.

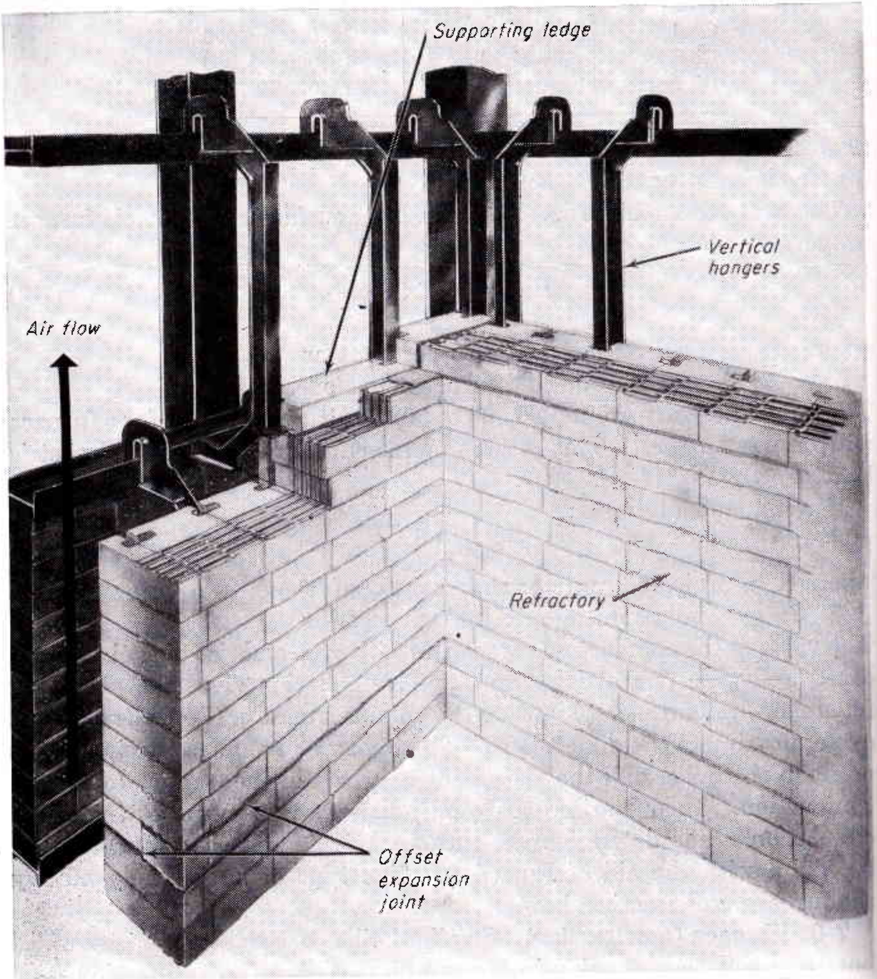
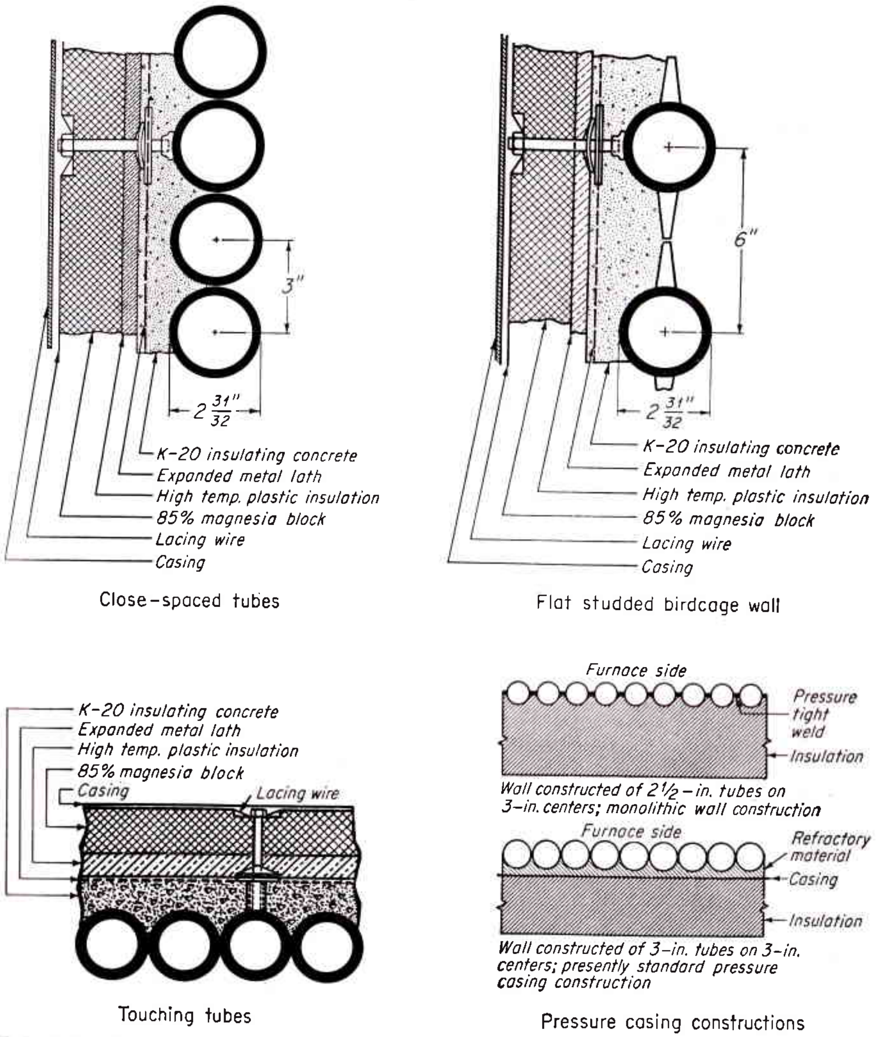


Fig. 7-20. One of many types of sectionally supported air-cooled walls.

Insulating layers reduce heat loss from the furnace but raise the refractory temperatures.

Low self-bearing walls need no steel support, but higher walls do. Steel ledges or hangers supported on steel columns or beams hold wall sections. Figure 7-20 shows a sectionally supported wall, the usual way also to build furnace arches.

Waterwalls have vertical tubes connecting lower and upper headers, which connect into the boiler circulation system. Furnaces may be totally or partly water-cooled. Figure 7-21 shows a series of waterwall arrangements; tube spacing, studs, and fins affect the heat transfer, wall slagging conditions, and furnace temperature.



Close-spaced tubes

Flat studded birdcage wall

Touching tubes

Pressure casing constructions

Fig. 7-21. Several designs of water-tube walls. (*Courtesy of The Babcock & Wilcox Company.*)

Steel plate or aluminum may be used for boiler casings. Pressurized furnaces have welded casings to make them gastight. A recent design uses a welded casing in contact with the wall tubes and backed up with insulation and an outer covering.

7-10. Furnace Bottoms and Refuse Removal. A stoker-fired furnace has the stoker as the major part of its bottom. A dump plate or ashpit occupies the remaining space. The smaller underfeed units have dump plates to receive the ash from the stoker extension grate. The larger units have ashpits at the tail of the extension grate.

In normal operation the ash builds up to a horizontal level at the stoker rear end. Clinker grinders extending the width of the stoker at the bottom of the ashpit control the depth of the ash. The grinder runs at intervals to break up large clinkers and pass the ash to a hopper or trough below. Water sprays usually cool the ash before it contacts the rolls.

All sizes of traveling-grate stokers have ashpits to receive the refuse dropping off the stoker rear. Grinders keep the ashpit level well below the stoker.

Pulverized-coal furnaces have either wet or dry bottoms. Ash from the fire settles down to the dry-bottom furnace in granular form. Furnace proportions ensure that the ash will be cooled to a solid before it falls into the hopper bottom. A water screen made up of widely spaced tubes prevents the full furnace radiation from melting the ash in the dry-bottom hopper. These tubes connect into the waterwall system. The ash falls between these tubes and is removed once a day or oftener.

In the wet-bottom furnace the furnace heat keeps the ash falling to and on the bottom in a molten state. The liquid ash drains through a hole to an exterior water-cooled pit. Wet bottoms find most acceptance today.

7-11. Steam-generator Ratings. Several methods are used to rate boilers; the smaller ones are specified in boiler horsepower,

$$\text{Rated boiler hp} = \frac{\text{heat-transfer area}}{10} \tag{7-5}$$

This term stems from an older day when reciprocating engines at the prevailing pressures developed about 1 hp for each 10 sq ft in the boiler supplying the steam. This practice has been superseded by giving boiler heat-transfer surface directly in square feet. Either total or itemized areas for the individual components, such as boiler tubes, waterwalls, superheater, economizer, and air, may be specific.

Older rating methods, slowly disappearing, were figured as

$$\text{Operating boiler hp} = \frac{\text{Btu per hr heat transfer}}{33{,}500} \tag{7-6}$$

$$\text{Per cent rating} = \frac{\text{operating boiler hp}}{\text{rated boiler hp}} \tag{7-7}$$

$$\text{Equivalent evaporation, lb per hr} = \frac{\text{heat transfer, Btu per hr}}{970} \tag{7-8}$$

These terms still apply to small boilers as standard practice.

Modern steam generators use the following methods of rating:

Steam generation = lb steam per hr with given steam pressure and temperature and feedwater temperature

Heat transfer = kilo (1000) or mega (1,000,000) Btu per hr

$$\text{kw rating} = \frac{\text{steam generation at rated capacity, lb per hr}}{\text{over-all station steam rate, lb per kwhr}}$$

The first expression is standard practice; the second is recommended by the ASME Standards Committee. The last has most practical use in figuring a plant or unit capacity for a central-station system.

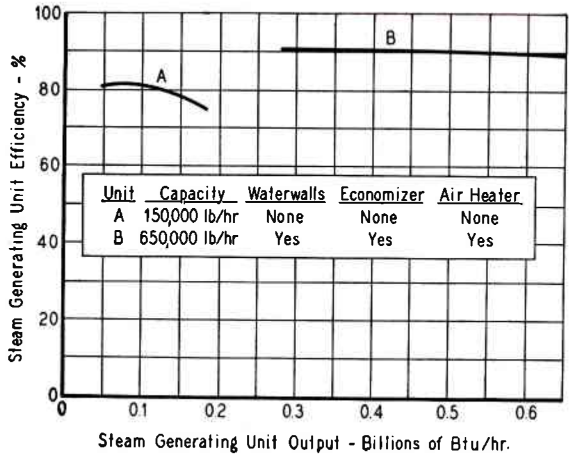


FIG. 7-22. Typical comparison of efficiencies of steam generators for units with and without extended heat-recovery surface.

Most boilers have both a continuous and a limited time peak rating. They can be overloaded by about 10 per cent for intervals of 1 to 4 hr depending on the fuel-ash characteristics.

7-12. Performance Characteristics. The range of losses in a steam-generating unit is outlined in Sec. 2-9. The losses depend on both the design of the unit and the rate of steam generation. Figure 7-22 shows the efficiency curves of two steam-generating units, one with 150,000 lb per hr steam capacity and having no waterwalls, economizer, or air heater, the other, a large unit, with 650,000 lb per hr steam capacity having complete waterwall cooling, economizer, and air heater. The comparison of the two curves shows the combined effect of size and use of heat-recovery components. The smaller boiler, despite the low efficiency, is of good design and more suitable for given conditions, all factors considered.

The operator's skill largely controls the operating efficiency of a

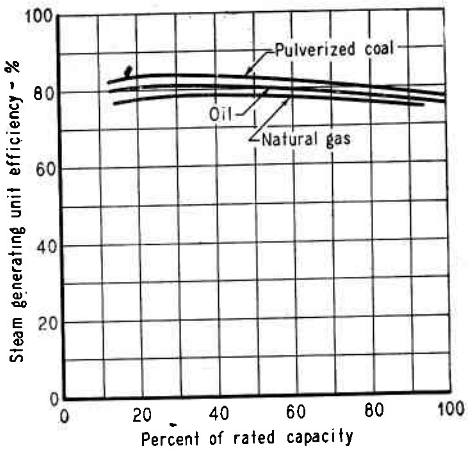


FIG. 7-23. Typical effect of changing fuels in a steam generator.

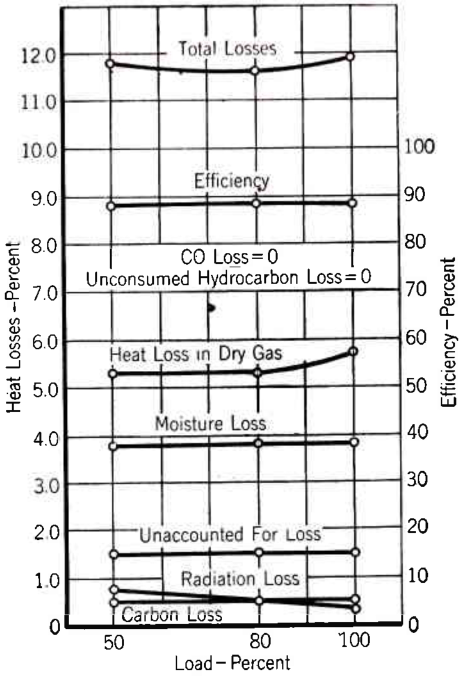


FIG. 7-24. Steam-generator losses and efficiencies (1,400 psia, 900-F steam, 300-F feedwater, 400,000 lb per hr, pulverized coal).

steam generator. Careless supervision may easily lower efficiency as much as 5 per cent. Heat-transfer surface fouling and too much excess air quickly reduce operating efficiency. Boiler, superheater, economizer, and air-heater surfaces must be cleaned at proper intervals.

The fuel type also controls boiler efficiency. Figure 7-23 compares the performance in burning bituminous coal, oil, or natural gas in the steam generator. The high hydrogen content of oil causes a higher water-vapor loss in the combustion gases and so reduces efficiency below that of coal. Gas suffers for the same reason and in addition transfers less heat because of its relatively nonluminous flame. Selecting a fuel depends principally on its availability and cost.

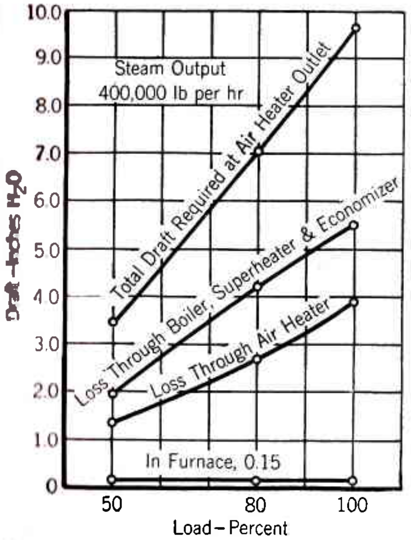


FIG. 7-25. Steam-generator pressure and draft variations.

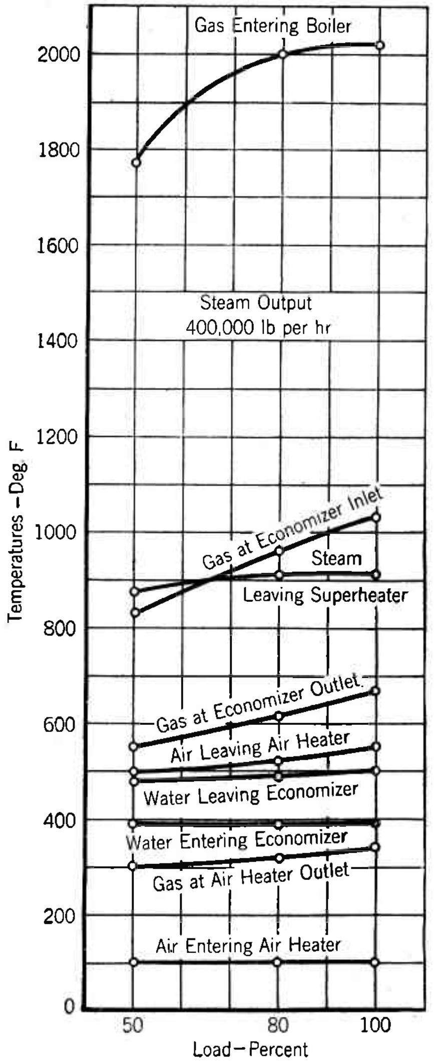


FIG. 7-26. Steam-generator steam-, water-, and gas-temperature variation.

Figures 7-24 to 7-26 give the complete performance of a steam generator. In addition to the efficiency, heat losses, drafts, and temperatures given in these figures the heat-release rates, excess air, and other factors may be plotted. These curves apply to a 400,000 lb per hr unit.

Combustion losses due to carbon monoxide are low or negligible, while the other heat-loss items vary slightly. The dry-gas loss increases with load owing to higher gas-exit temperatures and greater gas flows. The greater gas and air flows with increasing loads are reflected in the draft curves of Fig. 7-25. These indicate the increased losses with load increase as more air and gas flow through the resistance offered in the settings and heat-transfer surfaces, as in the economizer and air heater. In nonpressurized furnaces, furnace pressure stays slightly below atmospheric to prevent gas leakage through the casing.

The temperatures of air, gas, and water increase with load and fuel firing rate. Inlet air and leaving steam stay fairly constant.

PROBLEMS

7-1. A 78-in. HRT boiler is 14 ft long and has one hundred and twelve 3½-in. tubes. Calculate the total heating surface and the rated boiler horsepower.

7-2. A boiler has an effective heating surface of 1,650 sq ft. It has a capacity of 15,000 lb per hr on 280-F feedwater and 250 psia steam at 480 F.

Calculate:

a. Manufacturer's rated horsepower
b. Operating boiler horsepower
c. Percentage rating
d. Equivalent evaporation

7-3. The following data were obtained from a boiler test:

Heating surface, 15,200 sq ft
Steam pressure, 200 psig
Steam temperature, 500 F
Feedwater temperature, 230 F
Total feedwater, 850,000 lb
Time of test, 12 hr

Calculate:

a. Rated boiler horsepower
b. Operating boiler horsepower
c. Percentage rating
d. Equivalent evaporation

7-4. A 150,000-kw turbine uses 9.85 lb of steam per kilowatthour. Initial steam pressure is 350 psia and 700 F with feedwater at 278 F. Specify the following;

a. Maximum continuous capacity of boiler to supply this steam
b. Maximum 2-hr capacity if an overload of 10 per cent with a 15 per cent increase in turbine steam rate occurs

7-5. A gas-fired boiler delivers 300 lb of steam per hour of 95 per cent quality. Calculate the boiler horsepower and over-all efficiency for the following conditions:

Feedwater temperature, 60 F
Steam pressure, 50 psia
Barometric pressure, 30.2 in. Hg
Gas pressure, 4 in. H_2O
Gas temperature, 90 F
Gas consumption, 920 cu ft per hr
Heating value of gas, 540 Btu per cu ft at 60 F and 30.0 in. Hg

7-6. A steam-generating unit consisting of an economizer, a boiler, and a superheater receives 100,000 lb of water per hour at 750 psia and 260 F. Steam leaves the superheater at 650 psia and 600 F. The coal as fired has a heating value of 12,500 Btu per lb. Calculate the number of long tons fired per hour for a gross efficiency of 85.0 per cent.

7-7. The No. 5 boiler of the Ford Motor Company at River Rouge has a volume of 30,000 cu ft and is designed for a heat-release rate of 34,000 Btu per cu ft-hr when delivering 900,000 lb of steam per hour. Test data are as follows:

Evaporation, 901,000 lb per hr
Steam pressure, 1,329 psig
Steam temperature, 914 F
Feedwater temperature, 488 F
Over-all efficiency, 87.1 per cent
Heat value of coal, 13,810 Btu per lb

Compare the design and test values of the heat-release rates.

7-8. A furnace is to be designed for the following conditions:

Steam capacity maximum, 120,000 lb per hr
Over-all efficiency, 81.0 per cent
Steam pressure, 200 psia
Steam temperature, 420 F
Feedwater temperature, 210 F
Stoker projected grate surface, 300 sq ft

Determine the furnace height for a heat liberation not to exceed 40,000 Btu per cu ft-hr.

7-9. A coal has the following analysis:

$$\begin{array}{lll} C = 77.0 & N = 1.0 & A = 4.0 \\ H = 3.0 & S = 2.0 & W = 5.0 \\ O = 8.0 & & \end{array}$$

It is used under a boiler to generate 250 psia dry steam from feedwater at 190 F. If the boiler efficiency is 75.0 per cent, calculate the pounds of steam produced per pound of coal.

7-10. A 400-hp boiler operates at 200 per cent rating. Steam pressure is 250 psia and 500 F with feedwater at 250 F. Feedwater contains 8 grains per gal of salts. The boiler-water concentration is to be held to 150 grains per gal. Estimate the hourly blowoff in gallons.

7-11. A controlled-circulation unit delivers 350,000 lb of steam per hour at 720 psia and 750 F. Downcomers are 80 ft high, and a pressure drop of 60 psi occurs in the superheater. The circulating pump shows a discharge pressure of 875 psia at an efficiency of 72 per cent and is operated at a circulation ratio of 4.5:1.

Estimate the energy characteristic of the pump in:

a. Btu added per pound of circulated water
b. Horsepower

7-12. Data from a large twin-furnace public-utility steam generator are as follows:

Steam output, 1,850,000 lb per hr, 2,150 psia, 1010 F
Feedwater, 475 F
Fuel, bituminous coal; HHV 12,254 Btu per lb
Heat-release rate, 16,100 Btu per cu ft-hr

Heat-transfer area, sq ft:

Boiler, 5,300
Waterwalls, 32,160
Radiant superheater, 12,600
Convection superheater, 47,970
Economizer, 96,260
Air heater, 348,600
Reheater, 62,000

Projected furnace envelope, 33,340 sq ft
Furnace volume 154,300 cu ft

Compute:
(*a*) Hourly fuel rate
(*b*) Heat-release rate on area basis
Plot graph of heat-transfer area distribution.

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CHAPTER 8

SUPERHEATERS, REHEATERS, ECONOMIZERS, AND AIR HEATERS

8-1. Superheaters. These special heat-transfer surfaces, made up of parallel-tube circuits, receive steam from the boiler's release drum to raise its temperature. The heat absorbed from the furnace radiation or combustion gases first evaporates the moisture carried over with the steam and then superheats the steam to a predetermined level.

Figure 3-4 shows that superheating raises over-all cycle efficiency. A secondary advantage to superheating is the avoidance of too high a moisture in the last stages of the turbine. This must be limited to less than 12 per cent to avoid blade erosion. Superheat also reduces initial condensation losses in steam engines, raising their efficiency.

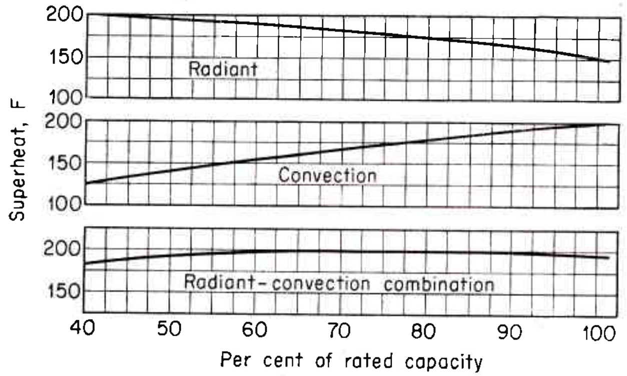

FIG. 8-1. Top curve shows drooping characteristics of radiant superheater, middle curve rising characteristic of convection type, and lower curve general result of combining two types. (*Courtesy of Combustion Engineering Company, Inc.*)

Superheaters usually have several tube circuits in parallel, with one or more return bends, connected between headers. Tube bends may be small or large radius, made by bending straight tubes, or may be forged and welded to tube ends. Tube ends are rolled or welded into the headers or passed through a detachable joint. Where high-temperature gases enter the superheater, spacing tubes widely reduces the deposition of molten ash.

Superheaters are classified as *radiant*, *convection*, or *combination*. Figure 8-1 shows how steam temperature changes with load for each of these types. The constant temperature of the combination type proves most desirable. Figure 8-2 shows superheater locations for smaller boilers, and Figs. 7-14 to 7-16 show the arrangement in larger steam generators.

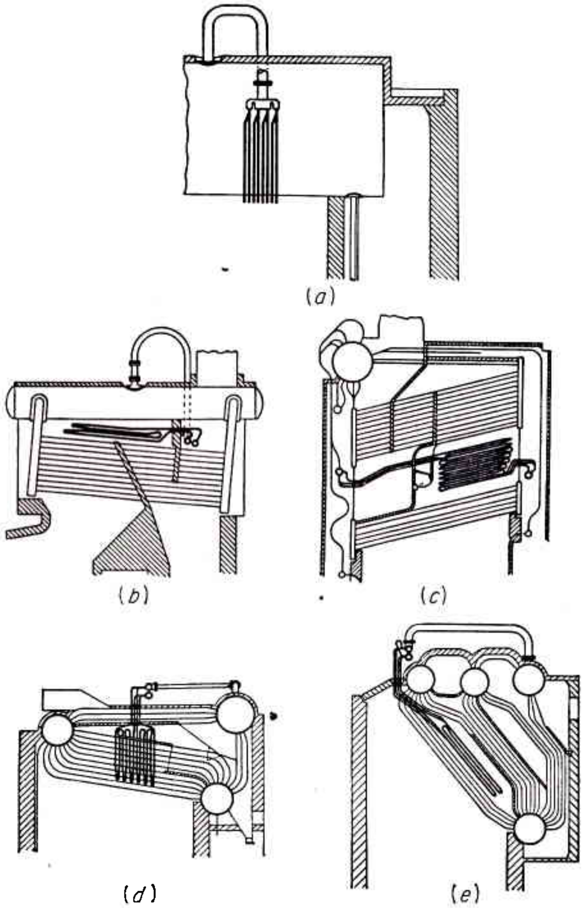

FIG. 8-2. Superheater classified by location. (*a*) Girth type for fire-tube HRT boiler; (*b*) overdeck; (*c*) interdeck; (*d*) intertube; (*e*) interbank.

Heat transfers from the combustion gas to the tube metal to the vapor in a superheater and works the metal at a high temperature (Fig. 4-10). Primary sections usually are arranged in counterflow, while parallel flow in secondary stages helps to reduce temperature stressing of the tube wall.

Superheater tubing gets rougher service than waterwall and convection tubes; therefore the metal must have high-temperature strength, high creep strength, and high resistance to oxidation. Special alloy steels serve this purpose.

Carbon steels are limited to about 950 F because of oxidation by higher-temperature steam. Some chromium-molybdenum alloys can be used to 1200 F. Tubes are normally smooth but may have extended surfaces.

As boiler steam pressures and temperatures rise, a greater proportion of the heat-transfer surface must be provided in the superheaters and reheaters (Fig. 8-3).

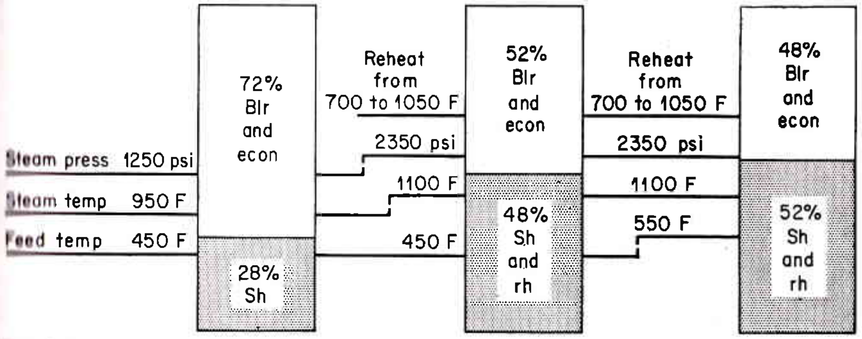

FIG. 8-3. Distribution of heat absorption in steam generator is significantly influenced by the pressure and temperature characteristics of the cycle used. Note increased amounts of heat transmitted in superheater *Sh* and resuperheater *rh* compared with boiler and economizer at higher pressures. (*Courtesy of The Babcock & Wilcox Company.*)

8-2. Superheat-temperature Control. Accurate steam-temperature control avoids overstressing superheater tubes and turbine front stages and maintains over-all cycle efficiency. Using both radiant and convection surfaces for superheaters holds the temperature nearly constant. This can be done by nesting a superheater in the first bank of boiler tubes or by having one part of the superheater on the furnace wall and the other part in the convection bank.

Figure 8-4 shows several methods used to control superheat precisely. These include:

1. Bypassing furnace gas around the superheater
2. Tilting burners in the furnace
3. Turning on or off auxiliary burners in the furnace
4. Desuperheater or attemperator using water spray or cooling coils
5. Precondensing control of saturated steam
6. Combustion-gas recirculation
7. Separately fired furnaces

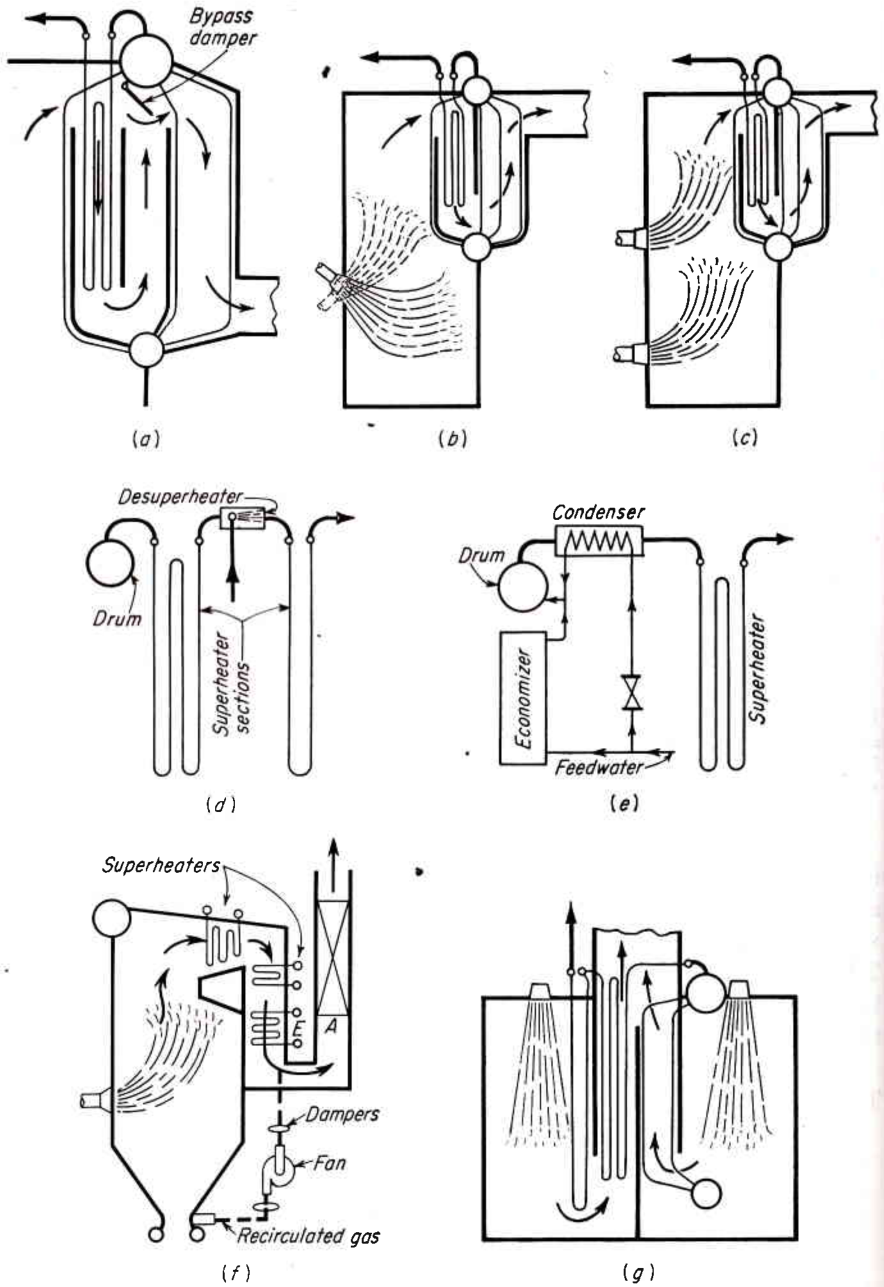

FIG. 8-4. Methods for superheat temperature control. (*a*) Gas bypass; (*b*) tilting burners; (*c*) auxiliary burners; (*d*) desuperheater (attemperator); (*e*) superheater condenser; (*f*) gas recirculation; (*g*) twin-furnace arrangement.

Choosing a convection superheater to give rated temperature at 70 per cent load makes it possible to hold constant temperature at higher loads by allowing part of the gas to short-circuit through a parallel pass.

In tilting burners downward in a furnace much of the heat will be absorbed by the waterwalls, and the gas entering the superheater will be *relatively* cool. Tilting the burners upward gives the waterwalls less chance of absorbing as much heat, and the gas entering the superheater is hotter, so raising the superheat. Burners tilt about 30° above and below the horizontal.

By turning burners, placed higher in the wall than the main bank, on and off we can control gas temperature into the superheater with the same effect as tilting burners. .

Steam temperature can be controlled by injecting water sprays either before the superheater or between sections of a superheater.

Forced recirculation of combustion gas from the economizer outlet to the furnace bottom acts like excess air and also blankets the furnace wall. This reduces waterwall absorption and raises superheater heat absorption.

The twin-furnace arrangement is an extension of the separately fired superheater. Varying the firing rate between furnaces controls superheat temperature. All these controls may be combined as needed.

8-3. Reheaters. Reheat cycles need two or more stages of superheating, one for turbine throttle steam and one or more for steam taken from the turbine for resuperheating. Early reheat cycles used separately fired superheaters or had heat exchangers using boiler water as the source.

Pressure drops in reheaters must be large enough to ensure good steam-flow distribution between parallel-tube circuits but small enough to avoid depressing cycle efficiency unduly. Reheat cycles inherently give better efficiency and have been increasingly adopted for central stations since 1945.

Modern units have reheaters in the same gas passages of the steam generator as the superheater. They may also be a combination type using both convection and radiant arrangements. The radiant surface often has the advantage of lowering the furnace-gas exit temperature enough to cool the ash below its fusion point. This prevents it from slagging the boiler convection surface. Figure 7-14 shows a modern reheat boiler.

Theoretically the best reheat pressures usually are lower than practical optimum pressures. Using higher pressures reduces the steam volume and tubing diameter and reduces the first cost. Intercept valves in reheat piping prevent overspeeding the turbine on a trip-out. These valves increase first cost and pressure losses in the piping.

8-4. Economizers. Combustion gases leaving the boiler convection surface carry considerable heat energy, at a temperature higher than the steam saturation temperature. Part of this energy can be recovered in an economizer (Fig. 8-5). The gases passing over the tube surface of an economizer heat the boiler feedwater flowing in the tube interior before the water enters the boiler drum.

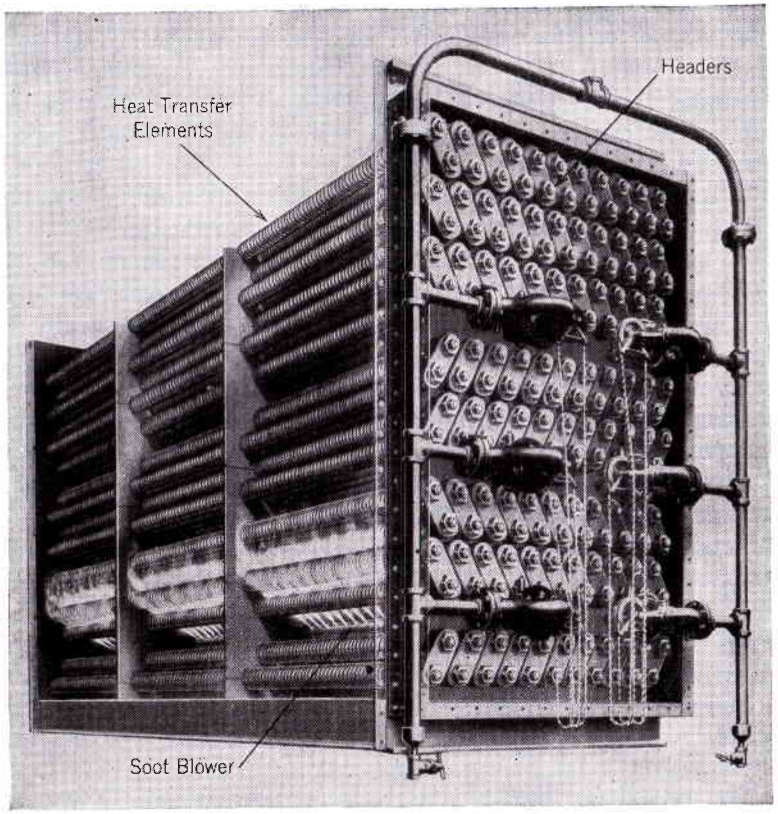

Fig. 8-5. Finned-tube economizer. Note headers outside of gas pass and finned extended-surface tubes. Soot blowers keep surfaces clean for better heat transfer. (*Courtesy of Foster Wheeler Corporation.*)

Steam-generator efficiency rises about 1 per cent for each 10-F rise in feedwater temperature produced by an economizer.

The justifiable cost for an economizer depends on the total gain in efficiency. In turn this depends on the gas temperature out of the boiler and the feedwater temperature to the boiler. Regenerative cycles inherently have high feedwater temperatures, and so the cycle must be studied carefully. Modern economizers have steel tubes to withstand high pressures and are integral with the steam generator. Economizers use small-diameter thin-walled tubes closely spaced. The feedwater temperature entering the economizer normally is kept above 175 F to avoid cooling the boiler gas to the dew point, which would cause severe corrosion with most fuels. Proper feedwater conditioning avoids internal corrosion.

One economizer type has parallel continuous loops welded to and running between a pair of water headers. Other types use various return-bend designs with horizontal tubes connected at their ends by welded or gasketed return bends outside the gas path. Gasketed bends may be box headers with handholes or flanged U ends. Figure 8-5 shows a staggered-tube economizer with headers. In the so-called "integral" economizer the feedwater enters a drum header and flows through tubes to a second drum, which discharges to the main boiler circulating system. The tube banks of this type of economizer stand at the outlet of the boiler convection passes.

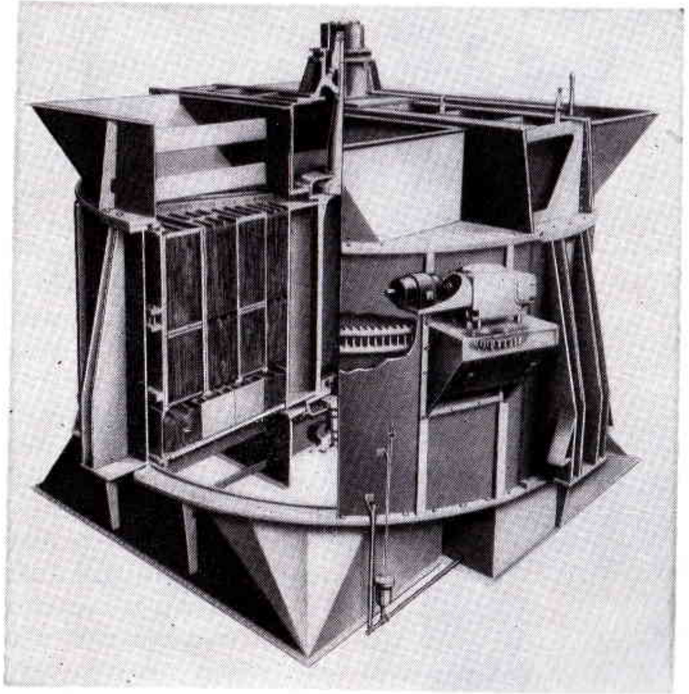

Fig. 8-6*a*. Regenerative air heater fits in limited space.

In almost all designs water flows back and forth across the gas path with counterflow heat exchange. Using small temperature drops, economizers need a relatively large surface. Economizer surface costs about as much as superheater surface, but much less than boiler surface.

An extended surface often proves economical; this may be cast-iron rings shrunk on steel tubes or steel fins welded lengthwise to the top and bottom of tubes.

8-5. Air Heaters. Incoming feedwater temperature sets a limit to the extent to which combustion gases can be cooled. Modern cycles produce about 400-F feedwater, meaning that exit-gas temperatures from an economizer will be higher. To recover this energy and preserve high cycle efficiency, an *air heater*, or *preheater*, may be used. In this heat exchanger the exit gases heat the incoming air for combustion.

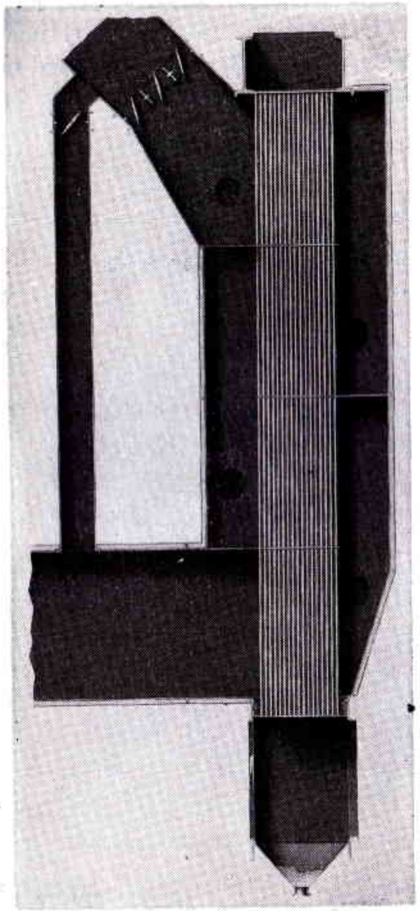

FIG. 8-6*b*. Tubular air heater passes gas through tubes; incoming air makes four crossflow passes over tubes.

Each 35- to 40-F drop in gas temperature raises steam-generator efficiency by about 1 per cent.

The recovered energy returns to the furnace and reduces the amount of heat that must be released by the fuel to maintain a given furnace temperature—i.e., it saves fuel. The average temperature throughout the steam-generator gas passes is higher and improves heat transfer. This gives a higher steaming capacity and decreases stack losses.

The amount of air heating used depends on the fuel and firing method. Pulverized coals use about 600 F temperature, especially if the coal is wet. With stokers air temperatures must be limited to about 250 F, or the air cannot cool the stoker enough to prevent distortion by expansion.

Figure 8-6 shows three types of air heaters: (1) regenerative, or rotary, (2) tubular, and (3) plate. The plate-type heater has narrow alternate lanes for gas and air passages. Baffles guide air direction for best heat-transfer-surface use.

In tubular air heaters the air flows over the tube exteriors and the gas through the interiors. Baffling leads the air in crossflow over the tubes. The tubes are welded or rolled into tube sheets at each end. In some cases tube-row staggering improves air distribution.

A regenerative air heater has a rotor made up of corrugated elements that have many narrow passages between them. Rotating at about 3 rpm, the rotor alternately passes through the leaving flue gas and the entering-air zones. The flue gas heats the metal rotor elements by flowing through the passages. The hot rotor, when in the air section of the heater, heats the air flowing through the rotor passages. In this way the rotor transfers heat from the gas to the air. Since an enormous amount of surface is provided by the elements, this makes a compact heat exchanger. Diaphragms and seals divide the unit diametrically, gas flowing on one side or zone and air on the other. The corrugated heating surface is made up in sections so that it may be easily removed for maintenance and replacement.

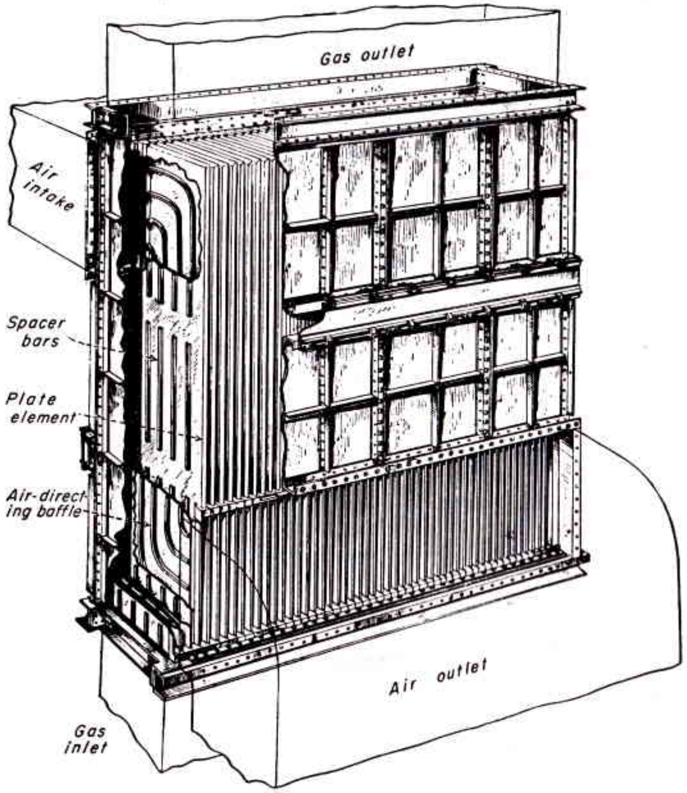

FIG. 8-6*c*. Plate-type air heater has air and gas flowing in alternate passageways.

Corrosion raises a serious hazard for air heaters. This can be avoided by limiting the cooling of the flue gas to a temperature above the dew point. This prevents moisture forming that can combine with the sulfur dioxide in the gas to generate an acid. Metal temperature can be kept above the danger point by bypassing some air around the heater or by recirculating air from the heater outlet to the forced-draft fan inlet. Current developments are trying ceramic-coated steel to protect against corrosion.

Air heaters must be periodically cleaned of fly ash to prevent clogging the gas passages. Fixed air or steam blowers prove adequate in many cases. Some stations find it necessary to wash the heating surfaces to remove ash.

8-6. Low-temperature Economizer. In the constant drive for high plant efficiency some modern designs use an economizer after the air heater in the gas circuit. The low-temperature economizer takes water only partly heated from the feedwater system and heats it by reducing gas temperatures below the dew point. The tubes are replaceable, and the heat saving is expected to more than pay for the occasional tube replacements needed because of corrosion. The heated feedwater returns to the feedwater system for further heating in higher-temperature bleed heaters.

PROBLEMS

8-1. Dry steam at 200 psia is supplied to a superheater in which 150 F of superheat occurs. The gas temperature drops 210 F in flowing over the superheater. Estimate the pounds of flue gas per pound of steam.

8-2. Steam at 650 psia and 700 F comes from a superheater receiving steam at a quality of 99 per cent. Feedwater enters the economizer at 300 F and 750 psia. What percentage of the heat transfer to the steam occurs in the superheater if there is a pressure drop through it of 30 psi?

8-3. A spray-type desuperheater is furnished with 120-F water. It is connected in a steam line carrying 350,000 lb per hr at 475 psia. Calculate the pounds per hour of water that must be sprayed in to maintain steam at 760 F when the boiler load causes steam to leave the superheater at 788 F.

8-4. A topping turbine receives steam at 1,200 psia and 900 F. Steam is exhausted at 260 psia and 600 F to two low-pressure units. A bypass in case of failure of the high-pressure unit consists of a pressure-reducing valve and a spray desuperheater using feedwater at 280 F. Calculate the water sprayed per hour if the two low-pressure turbines receive 200,000 lb of steam per hour when the high-pressure unit drops load.

8-5. Test data show the following gas analyses before and after the air heater:

Table P-1

	Entering	Leaving
CO_2	9.7	9.2
O_2	4.0	4.9
N_2	86.3	85.9

The coal used shows a carbon percentage of 72.0 per cent. Estimate the air leakage in the air heater per pound of coal fired.

8-6. A power station is operated on a reheat cycle under the following conditions:

Steam from superheater, 1,200 psia and 820 F
Feedwater, 1,500 psia and 350 F
Steam into reheater, 260 psia and 540 F
Steam out of reheater, 250 psia and 820 F

If 95 per cent of the steam flow is reheated, what percentage of the total heat transfer to the steam occurs in the reheater?

8-7. A 165,000-kw turbine uses 8.7 lb of steam per kilowatthour. Steam pressure is 625 psia and 700 F with feedwater at 320 F. The steam-generator efficiency is 88.0 per cent. A gas-heated reheater is used taking saturated steam from the first bleed point of the turbine at 400 psia and reheating to the original temperature. Steam to the high-temperature feedwater heater is 8.0 per cent of the turbine input. Calculate the pounds of coal used per hour based on a heating value of 13,280 Btu per lb.

8-8. A modern steam generator delivers 700,000 lb per hr of steam at 1,800 psia and 1000 F. Feedwater is supplied at 450 F and 2,100 psia. Pressure drops 100 psi through the superheater. Coal with an analysis $C = 67\%$, $H = 5\%$, $O = 15\%$, $N = 2\%$, $S = 3\%$, $A = 8\%$, $HHV = 12{,}420$ Btu per lb is pulverized and burned with 22 per cent excess air. A counterflow convection superheater is used, and the combustion gas is cooled to within 120 F of incoming steam temperature. Estimate the gas temperature leaving the furnace. Make reasonable assumptions for any added data required.

8-9. The following data apply to a steam generator with economizer and air heater:

Steam output, 185,000 lb per hr
Blowdown, 3 per cent
Gas flow, 240,000 lb per hr
Gas temperature to economizer, 770 F
Gas temperature leaving air heater, 300 F
Air flow, 188,000 lb per hr
Water temperature to economizer, 410 F
Water temperature from economizer, 440 F
Air temperature to air heater, 110 F
Economizer, 3,400 sq ft
Air heater, 46,000 sq ft

Compute, making needed rational assumptions:
(*a*) Gas temperature leaving economizer
(*b*) Air temperature leaving air heater
(*c*) Heat-transfer coefficients for both economizer and air heater

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CHAPTER 9

DRAFT SYSTEMS

9-1. Combustion Needs. The combustion process in a furnace can take place only when it receives a steady flow of air and has the combustion gases constantly removed. The steam-generator draft system induces this air and gas flow. When only a chimney is used, the system is a *natural-draft* system; when this is augmented with a forced-draft (f-d) or induced-draft (i-d) fan or both, the system is a *mechanical-draft* system.

Small boilers use natural draft, but large units need mechanical draft to move the large volumes of air and gas against the flow resistance. Chimneys or stacks contribute only a small draft to the total needed in the large units. They help to discharge gases and fly ash high enough above ground to dilute them with air and minimize the air-pollution nuisance. Maximum economical heights are of the order of 200 ft and less.

The net movement of combustion gases in a steam generator is vertically upward as they flow from furnace to convection passes to air heater to stack. The hot gases in these passages weigh less than an equal column of outside cold air. This causes a pressure difference that moves a flow of air and gas steadily through the fireside of the steam generator. The maximum pressure difference, or *static* stack draft (chimney ± other vertical passages), available to create flow is

$$D_s = \frac{12}{62.4} h_s(\rho_a - \rho_s) \tag{9-1}$$

where D_s = stack draft (maximum pressure difference created), in. H_2O

h_s = stack height, ft (including vertical gas-pass height)

ρ_a = air density, lb per cu ft (depends on barometric pressure and air temperature)

ρ_s = gas density, lb per cu ft (depends on gas pressure and temperature)

In computing stack effect for an arrangement of more than one vertical gas passage in series use Eq. (9-1) for each vertical separate section. Gas density varies considerably from section to section,

$$D_s \text{ for series} = \Sigma D_s \text{ of sections} \tag{9-2}$$

Use minus (−) values for D_s per section when gas flows downward in the section and plus (+) values per section when gas flows upward.

Chimneys are generally made of brick, steel plate, or concrete. Steel and concrete stacks may be lined for heat resistance or left plain.

Draft-system designers use the following terminology:

Draft is the difference between absolute gas pressure at any point in a gas-flow passage (furnace, chimney, air heater, etc.) and the ambient (same elevation) atmospheric pressure. Draft is plus if $P_{\text{atm}} < P_{\text{gas}}$ and is minus if $P_{\text{atm}} > P_{\text{gas}}$. Alternatively values are called "pressure" and "draft," respectively, with no algebraic sign being used. Units are inches of water as measured by a U tube or inclined manometer.

Draft loss is the pressure loss caused by friction between two points in the gas-flow path.

Draft differential is the pressure difference between two points in a gas flow (algebraic sum of draft loss and stack effect).

9-2. Draft-system Flow Resistances. The flow pressure pattern in a draft system may be evaluated by

$$D_F + D_S = D_A + D_G + D_V \tag{9-3}$$

where D_F = total fan-effect pressure

D_S = net stack effect (chimney ± vertical passages)

D_A = draft pressure loss on air side

D_G = draft pressure loss on gas side

D_V = gas-exit velocity pressure

In this equation D_A is the sum of friction losses in air ducts, bends, air heater, stoker bed, secondary-air pressure at fuel burner, etc. D_G is the sum of friction losses in gas ducts, bends, economizer, air heater, superheater, boiler setting, chimney, etc.

Chimney and duct losses may be calculated by equations like (4-51) and empirical modifications. Manufacturers supply data for equipment like air heaters, economizers, boiler passes, superheaters, etc. These losses follow a parabolic law,

$$D_1 = D_2 \left(\frac{W_{s1}}{W_{s2}}\right)^{1.8 \text{ to } 2.0} \tag{9-4}$$

where D is the draft loss at rating W_s.

Fuel-bed or stoker losses depend on fuel size, bed thickness, and combustion rate. Manufacturers supply data on these as needed. Figure 9-1 shows the principal parts of a draft system designed for negative furnace pressure, usually called *balanced draft*. The draft steadily drops from f-d fan outlet to i-d fan inlet.

Table 9-1 gives a specific draft performance for the steam generator shown in Fig. 7-15.

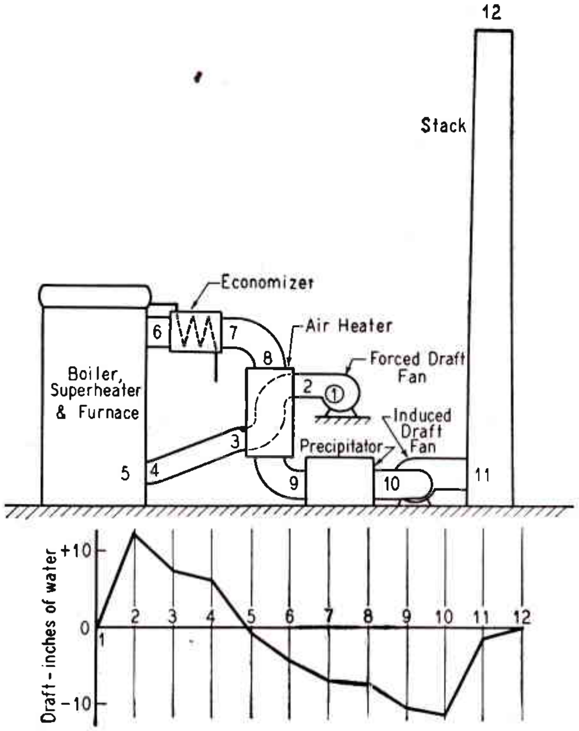

FIG. 9-1. Draft system of large steam generator and typical draft variation at rated capacity. Note *balanced draft* at furnace, point 5.

TABLE 9-1. TYPICAL DRAFT LOSSES[a]

Union Electric Company of Missouri-Meramec Station, St. Louis, Mo.
Load, 1,850,000 lb per hr Fuel, coal
Pressure, 2,150 psig Temperature, 1010 F

Component	*Draft values, in.* H_2O
Draft in furnace	0.10
Draft loss through boiler	0.10
Draft loss through superheater and reheater	2.50
Draft loss through economizer	1.70
Draft loss through air heater	2.30
Draft loss through flues	0.65
Draft loss, total	7.35
Air pressure, loss through air heater	2.85
Air pressure, loss through ducts	0.83
Air pressure in wind box	4.60
Air pressure, loss steam coils	0.40
Air pressure, total	8.68

[a] Courtesy of Foster Wheeler Corporation.

9-3. Fan Characteristics. Today's large boilers with capacities of almost 4 million lb steam per hour would be impossible to run without the aid of draft fans. A chimney of any reasonable height would be incapable of developing enough draft to move the tremendous volume of air and gases. Fans reduce the height of stacks needed for adequate draft.

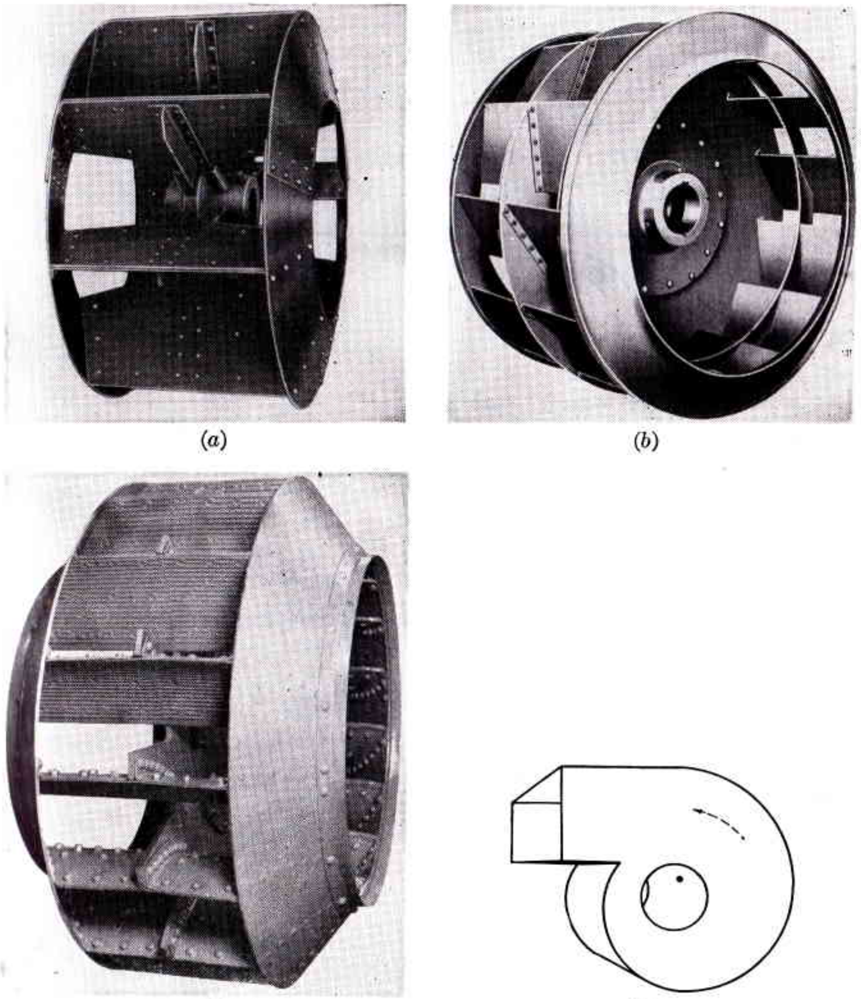
(*a*) (*b*)
(*c*) (*d*)

FIG. 9-2. (*a*) Steel-plate fan rotor; (*b*) backward-curved-blade fan rotor; (*c*) forward-curved-blade rotor; (*d*) fan scroll housing.

Fans have a rotating element (rotor) and a stationary housing (casing) as in Fig. 9-2. Fan performance depends on the shape of the blades

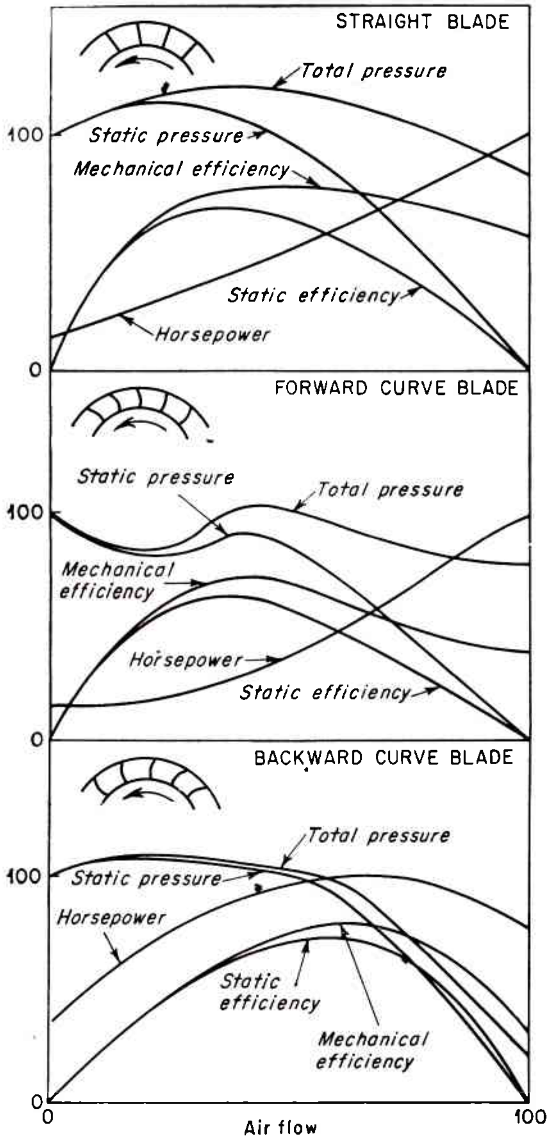

FIG. 9-3. Basic fan performance characteristics at *constant rotor speed.* Fans always operate on their characteristics.

in the rotor, they are classified as: (1) straight, or radial; (2) backward-curved; (3) forward-curved; and (4) double-curved.

Figure 9-3 compares the performance curves on a qualitative basis. Manufacturers supply such data for each type of fan they offer. These curves define constant-speed performance. With speed change the theoretical performance at constant efficiency follows the fan laws,

$$\text{cfm} \propto \text{rpm} \qquad \text{Total head} \propto \text{rpm}^2 \qquad \text{hp} \propto \text{rpm}^3$$

Backward-curved blades have air leaving at low velocity from the blade tips, making them suitable for high rotor speed. Their power consumption reaches a maximum at an intermediate flow so that the fan driver cannot be overloaded. Power plants ordinarily use this type of fan for forced draft. The forward- or straight-bladed fans are preferred for i-d fans since their lower speed retards blade erosion caused by fly ash.

In the curves of Fig. 9-3 static efficiency is defined as

$$\eta_s = \frac{W_i}{W_a} \times 100 \tag{9-5}$$

where η_s = static efficiency, %
W_a = actual input to fan, hp
W_i = input to ideal equivalent fan, hp (static air horsepower)

$$W_i = V \times \rho_a \times \frac{h_s}{33{,}000} \tag{9-6}$$

where V = input volume to fan, cfm
ρ_a = input fluid density, lb per cu ft
h_s = static pressure developed, ft of fluid flowing

NOTE: If in Eq. (9-6) h_s is replaced by the total head developed (velocity + static head), the result is the *total air horsepower.*

9-4. Fan Selection. Figure 9-4 shows the relative positions of the f-d and i-d fans in the steam-generator draft system. Fan selection depends on evaluating the *volume* flows needed at design conditions and the *pressure differences* that will establish these flows. Some excess often will be provided for to allow for fuel variation and possible leakage.

Forced-draft Fans. These fans must handle maximum anticipated flow based on theoretical and excess air needs. Any leakage or infiltration as in Fig. 9-4 may seriously alter the load on the fan. Volume also depends on air-intake temperature, which varies with fan location and seasons, and on barometric pressure.

The boiler designer figures the air volume needed on the basis above and allows an additional 20 to 25 per cent capacity to allow for unforeseen contingencies. Pressures depend on all the resistances on the air side of the furnace [D_A of Eq. (9-3)] and the possible stack effects in the air heaters and ducts. Factors such as coal size, caking fuel beds, slagging furnaces, and gas passes make it difficult to predict exact pressure losses.

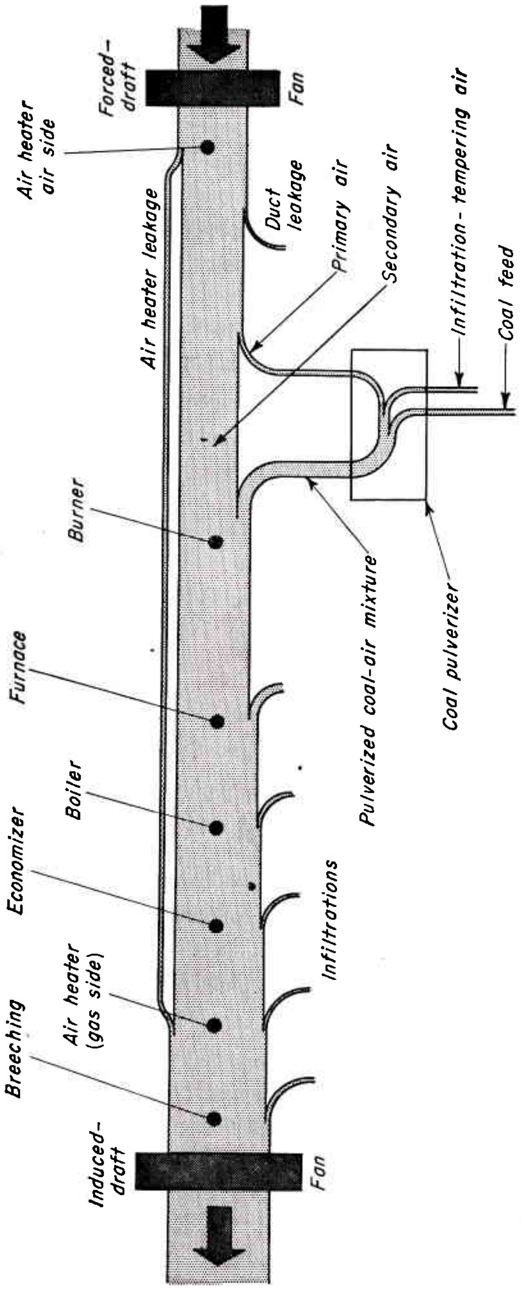

FIG. 9-4. Location of f-d and i-d fans in a draft system. Note leakages and infiltrations, which affect fan selection.

Pressure excesses of 40 to 50 per cent for resistance variation are often used in selecting fans. The f-d fan must be reliable and efficient and must have a self-limiting power input to prevent overload if discharge backpressure should suddenly decrease to atmospheric. Forced-draft fans handle relatively clean air at high-speed rotation and usually are fitted with backward-curved blades.

Induced-draft Fans. A fan of this type (Fig. 9-4) usually works at the outlet point of the steam generator after the dust collector. It handles relatively high-temperature gas with some dust loading. Gas volume depends on many factors: air supplied, fuel type, infiltration air, gas pressure and temperature, and the moisture content.

Dust content affects volume slightly but more markedly affects gas density, reflected in static pressure. Volume calculations are usually uprated by 15 to 20 per cent to account for variables.

Induced-draft fans usually have a negative static pressure at inlet and outlet; when discharge-control dampers or long breechings are used, the outlet pressure may be positive. The static pressure developed depends on resistance [D_G, Eq. (9-3)] between fan inlet and furnace plus any stack effect in the furnace, economizer, or air heater. Good practice assumes some 20 to 30 per cent excess pressure above calculations. The i-d fan handles hot and dust-laden gas; designers use almost any of the basic types, but the straight or forward-curved types with lower rotor speeds help reduce maintenance.

Forward-curved blades are used for moderate pressures and the straight, or radial, type for higher pressures. Induced-draft fans have water-cooled bearings, and the casing and blades may have renewable strips at the points that erode most quickly.

Figure 9-5 shows the performance of a variable-speed centrifugal fan discharging to a duct system. The intersections of system resistance and speed curves show the operating points. For any desired volume flow the speed must be adjusted to match the system resistance at that flow. For constant-speed operation an artificial or variable resistance like a damper must be introduced into the system to control the flow. This method reduces operating efficiency but is a low-cost control.

9-5. Fan Drives and Control. Directly connected motors drive f-d and i-d fans in many plants. Alternating-current motors ordinarily are used in modern power plants; since speed control with slip-ring motors or multiple-winding induction motors is expensive, this system of speed control is seldom used. Most plants use inlet louvers or discharge dampers to control flow with constant-speed motors.

Some designers prefer to use constant-speed motors but vary fan rotor speed by variable-ratio hydraulic or magnetic couplings. The constant motor speeds selected vary from 1,200 to 1,800 rpm for f-d fan drives and 720 to 1,200 rpm for i-d fan drives.

9-6. Pressurized Furnaces. Many of the newest large steam generators have f-d fans only. These furnaces run under positive pressure and must be enclosed in a gastight casing. This raises boiler efficiency slightly by eliminating leakage air and infiltration. Investment in the i-d fan is eliminated, but this is offset somewhat by the cost of the pressure-tight casing. Maintenance and operating energy of the i-d fan are eliminated at the expense of somewhat higher f-d-fan energy input.

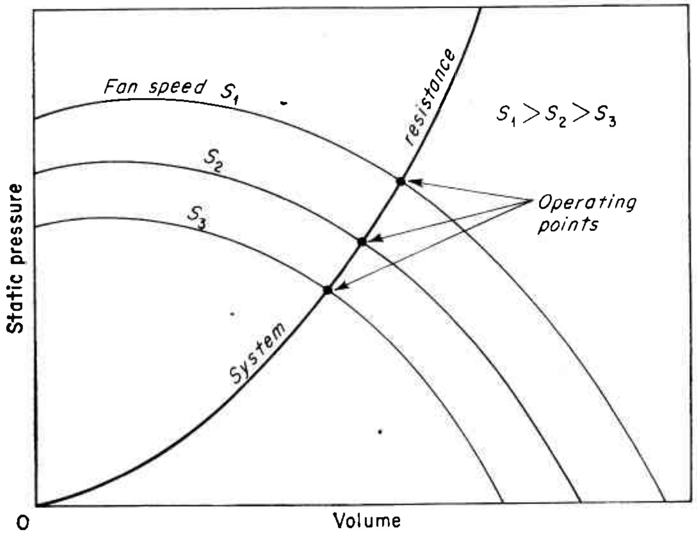

FIG. 9-5. Performance of variable-speed fan connected to a duct system. Intersection of speed and system resistance curves determines operating point.

9-7 Auxiliary Fans. An *overfire-air fan* may be used with stokers to inject air above the fuel bed to complete combustion and aid mixing. These fans may introduce about 10 per cent of the total combustion air.

Coal pulverizers use a pulverizer exhauster fan to remove the coal-and-air mixture from the unit. Pulverizers use air-to-coal ratios by weight of 1.5:1 to as high as 3:1 at temperatures of 150 to 200 F. These fans are usually integral with the pulverizer. A *vent fan* supplies heated air or gas to dry the coal in a pulverizer.

PROBLEMS

9-1. Flue gas flows through three vertical passages *A*, *B*, and *C* in series in that order. Data are as follows:

TABLE P-1

Passages	Element	Height, ft	Flow	Temperature, °F
A	Economizer	60	Up	1550
B	Air heater	130	Down	1100
C	Chimney	150	Up	280

Assume a gas density of 0.0418 lb per cu ft at 500 F and an air density of 0.0768 lb per cu ft at 60 F. Calculate the draft at the entrance to passage *A* due to stack effect.

9-2. A steam generator has two f-d fans in parallel with a capacity of 150,000 cfm each. Air inlet is at 110 F, a static pressure of 10.5 in. H_2O, is developed, and the fan speed is 1,220 rpm. (*a*) Based on a fan input of 345 hp each, calculate the static efficiency. (*b*) Calculate the capacity, head, and power requirements for a speed increase of 15 per cent.

9-3. Compare the fan power needed for a conventional and pressure-setting furnace, based on data as follows:

TABLE P-2

	Air flow, lb per hr	Gas flow, lb per hr	Static pressure, in. H_2O	Air temperature, °F	Gas temperature, °F
Conventional........	371,855	431,255	7.20 induced draft; 8.12 forced draft	90	375
Pressure furnace.....	371,855	....	14.18 forced draft	90	375

Assume a static fan efficiency of 72 per cent in all cases and a gas density of 0.0708 lb per cu ft at 60 F.

9-4. Determine the volume and pressure specifications for selecting an f-d fan for the following installation:

Steam generator:
  Load, 150,000 lb per hr
  Pressure, 850 psia
  Steam temperature, 900 F
  Feedwater temperature, 375 F
  Efficiency, 88.2 per cent
Air-heater resistance at 60,000 lb per hr 0.7 in. H_2O
Unlined steel duct, 75 ft
Pressure in secondary-air burner box, 4.3 in. H_2O

Make rational assumptions when necessary, and employ reasonable excess factors.

9-5. Compute the required motor capacity needed for the f-d fan under the following conditions:

Coal rate, 10 long tons per hr
Coal analysis:

$C = 78.0 \quad H = 3.0 \quad O = 3.0$
$S = 1.0 \quad A = 8.0 \quad W = 7.0$

Excess air, 30 per cent
Plenum chamber pressure, 7 in. H_2O
Mechanical efficiency of fan, 60 per cent

9-6. The draft system of a boiler consists of a stack and f-d fan. A 200-sq ft traveling-grate stoker is employed, burning anthracite coal. Estimate the stack height and fan motor capacity for the following data:

Flue-gas temperature, 500 F
Inlet air, 14.7 psia and 75 F
Coal heating value, 12,800 Btu per lb
Excess air, 25 per cent with buckwheat No. 2 coal
Combustion rate, 35 lb per sq ft per hr
Stack friction drop plus velocity pressure, 0.042 in. H_2O
Boiler and breeching loss, 0.65 in. H_2O
Mechanical efficiency of fan, 70 per cent

9-7. Compute the head that an i-d fan must develop for the following conditions:

Maximum boiler load, 250,000 lb per hr
Draft loss in duct work, 0.3 in. H_2O at 250,000 lb per hr
Air-heater draft loss, 2.0 in. H_2O at 120,000 lb per hr
Boiler draft loss, 1.1 in. H_2O at 200,000 lb per hr

Assume that losses vary as the 1.8 power of the steam flow.

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CHAPTER 10

FUEL AND ASH HANDLING

10-1. Coal and Ash Handling. Coal originates in veins underground and is claimed by deep mining, strip mining, or auger mining. Rail, boat, or truck transport brings the coal to a receiving point in the power plant. Mechanical systems in the plant move the coal to storage or to the furnace. At the end of the combustion process other systems remove the ash from the furnace to the point of disposal in the plant.

The capacity of a fuel-handling system is usually several times the plant fuel-burning rate. This allows replenishing the coal for a 2-day supply in a single shift of 8 hr, or in two shifts for very large plants, and so holds down labor and demurrage costs. Bunkers directly supplying the furnaces usually hold enough coal for 1 day's operation. Significant factors considered in selecting fuel-handling systems are: (1) plant fuel rate, (2) plant location in respect to fuel shipping, and (3) storage area available.

10-2. Handling Systems. No system can be termed typical. Figure 10-1 shows the various steps in coal handling and the equipment used. Several plant arrangements can be indicated. For instance, a simple heating plant in an office building may receive coal by truck and have it dumped into a bin. A screw conveyer then takes the coal to a stoker hopper, an arrangement shown by sequence *A-F-G-I*.

A small sized plant may receive coal by rail to a track hopper that feeds to a skip hoist. The hoist lifts the coal to a bunker that feeds a weigh larry which delivers the coal to a stoker hopper (sequence *A-D-F-H-I*).

A larger plant may receive coal by barge; an unloading tower empties the barge to a feeder that discharges to a crusher. From the crusher the coal goes to a belt conveyer with a tramp-iron magnetic separator. This belt lifts the coal to the bunker tops, where crossbelts and trippers distribute the coal to the bunkers. From here chutes deliver the coal to weigh scales and then to the pulverizer (sequence *A-B-C-D-F-G-H-I*).

When coal is stored outside, it would be bypassed, after leaving the crusher, to chutes that discharge to carryalls for outdoor storage (sequence *A-B-C-D-E*, a stock-piling operation). These few examples show some of the equipment combinations used in practice.

10-3. Elements of Mechanical Systems. Rail cars delivering coal must be spotted exactly at the unloading point. This may be a trestle over a track hopper or a rotary car dumper (Fig. 10-2*a*). Wet and frozen coal needs added handling. Car shakers, loosening augers (accelerators), or displacers free the coal and may even be used to speed up normal handling. Cars may be thawed in heating sheds or by standing over oil-fired thawing pits.

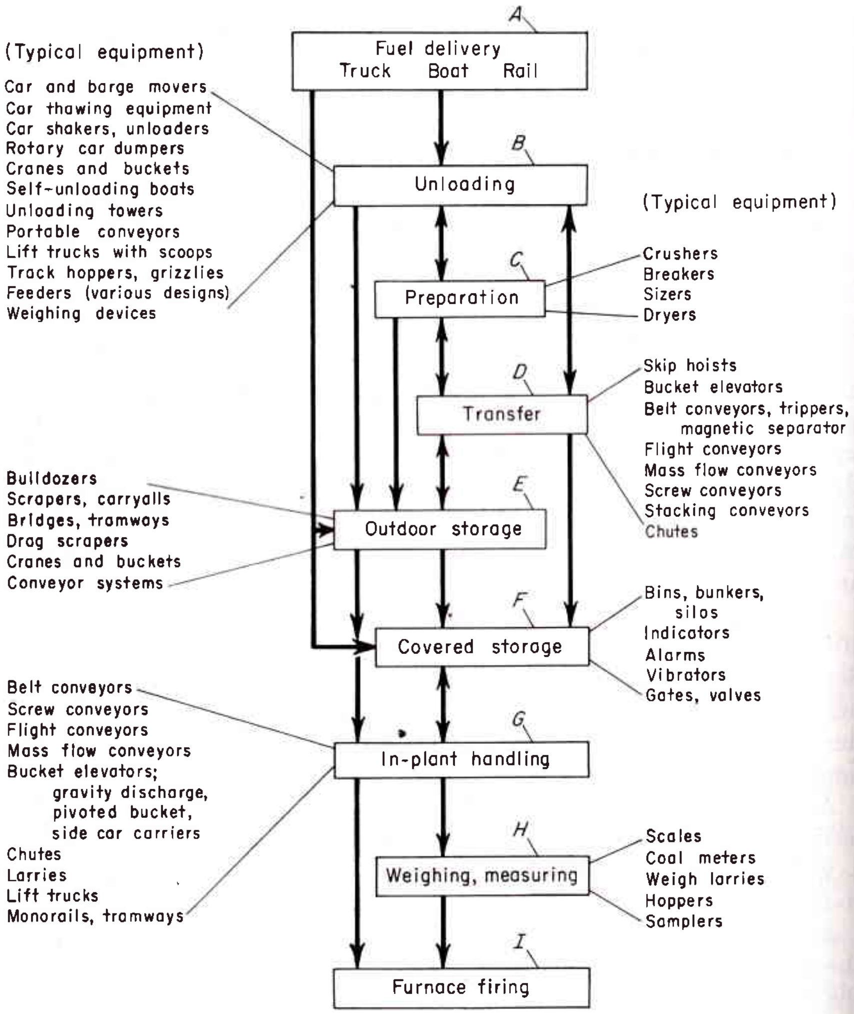


FIG. 10-1. Flow chart of basic steps in coal-handling system. Typical devices used for each function are listed. A single device frequently serves several functions.

Coal towers (Fig. 10-2*b*) or unloading bridges remove coal from barges in large operations. Buckets raise the coal from the positioned barge to a hopper feeding a conveyer. Self-unloading barges are used on the Great Lakes. A feeder (Fig. 10-3*a*) removes the coal discharged from the hoppers to conveyers to reduce wear. Receiving coal at a uniform rate makes conveyers more effective. Feeders may also supply crushers at the unloading point in some systems. Feeders run horizontally or

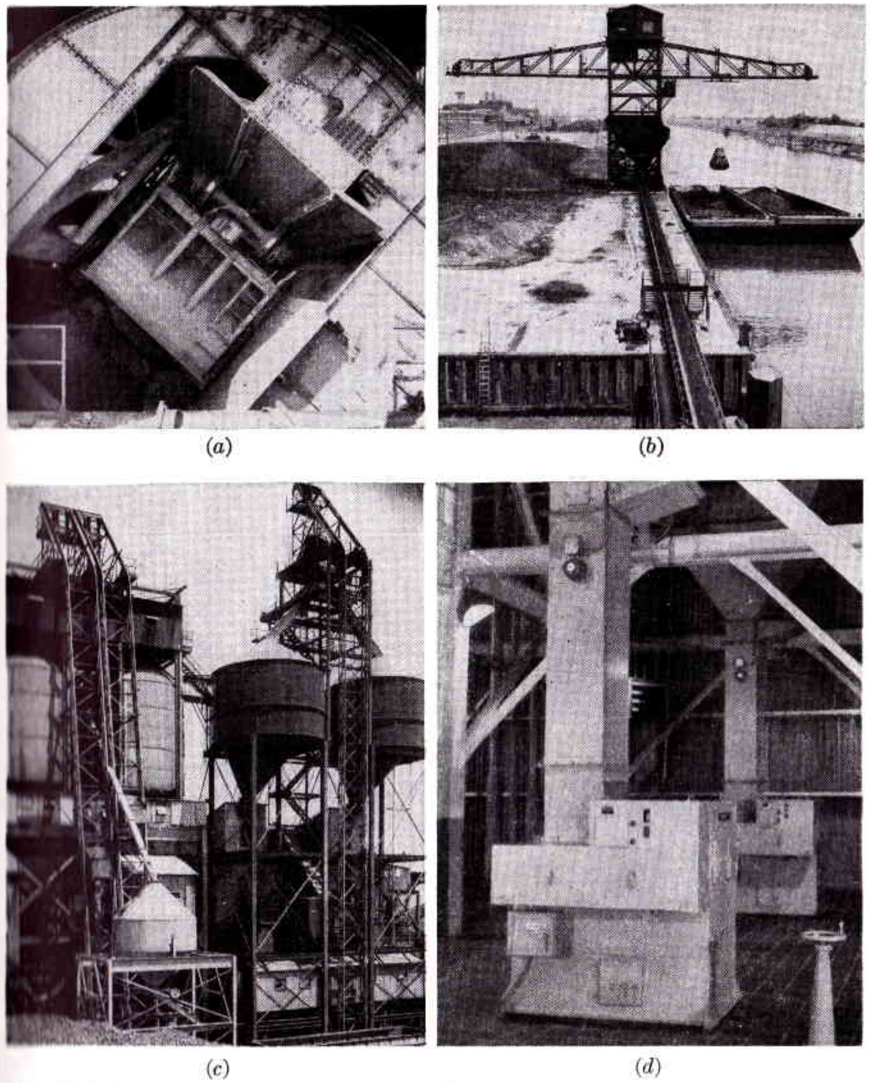

(*a*) (*b*)
(*c*) (*d*)

FIG. 10-2. Coal-handling components. (*a*) Rotary car dumper; (*b*) trolley tower with bucket unloader; (*c*) skip hoist; (*d*) coal scale. [(*a*), (*b*), *and* (*c*) *Courtesy of Link-Belt Company.* (*d*) *Courtesy of Richardson Scale Company.*]

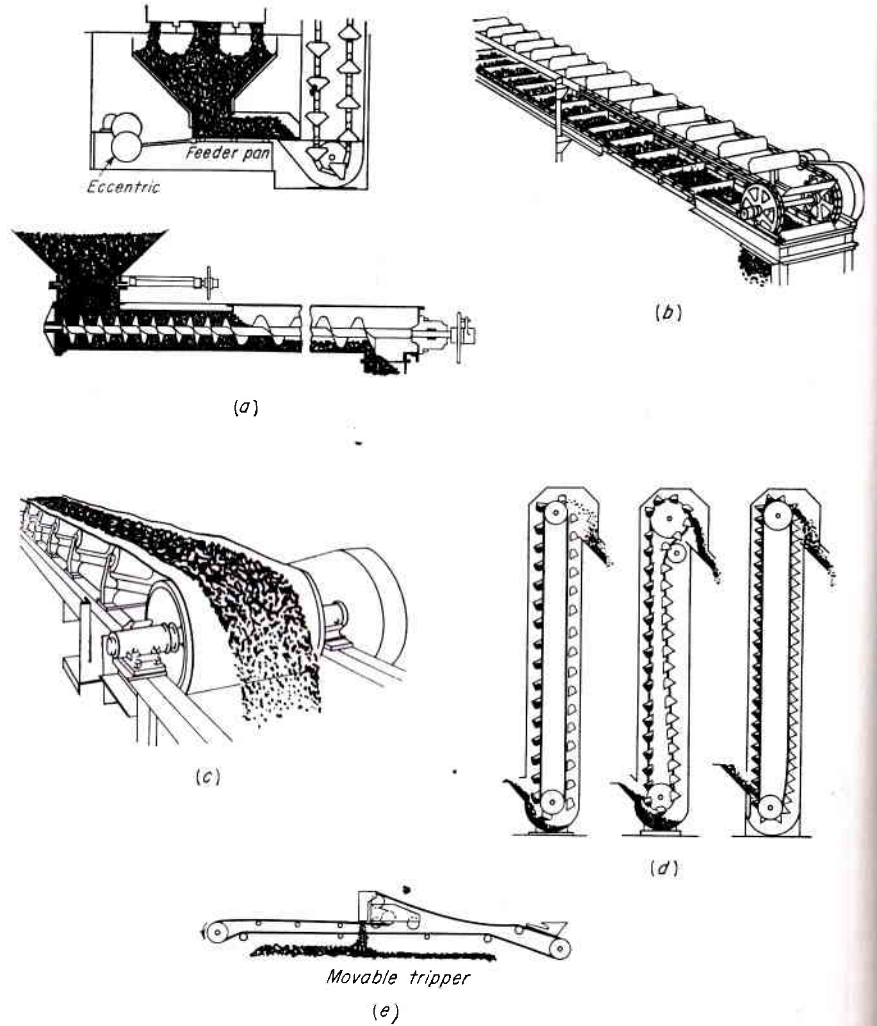


FIG. 10-3. Examples of unloading and transfer equipment. (*a*) Feeders (top, reciprocating; bottom, screw); (*b*) flight conveyer; (*c*) belt conveyer; (*d*) end-discharge belt conveyer; (*e*) movable belt tripper. (*Courtesy of Power.*)

at a slight angle from the hopper bottom. Trucks usually dump coal to outside storage or to a bin below road level.

Coal is stored in bins or silos in plants and also in outdoor areas. Conveyers deliver coal from the unloading point to storage, or to the furnace, and from storage to the firing equipment (Fig. 10-3*a* to *d*).

Skip hoists and bucket elevators lift coal vertically, while belts and flight conveyers move the coal horizontally or on inclines. Coal usually discharges from the ends of belt conveyers. But in discharging coal to a long bunker or a series of silos, belts have trippers that can dump the coal anywhere along the belt travel. One design (Fig. 10-3*e*) passes the belt over rollers that bend it to an S shape. The coal spills into a hopper that leads it by two side chutes into the space below the belt. The carriage carrying the pulleys and hoppers can move the full length of the belt travel on rails. Permanent magnets or electromagnets in pulleys remove scrap iron from the coal as it discharges from the belt.

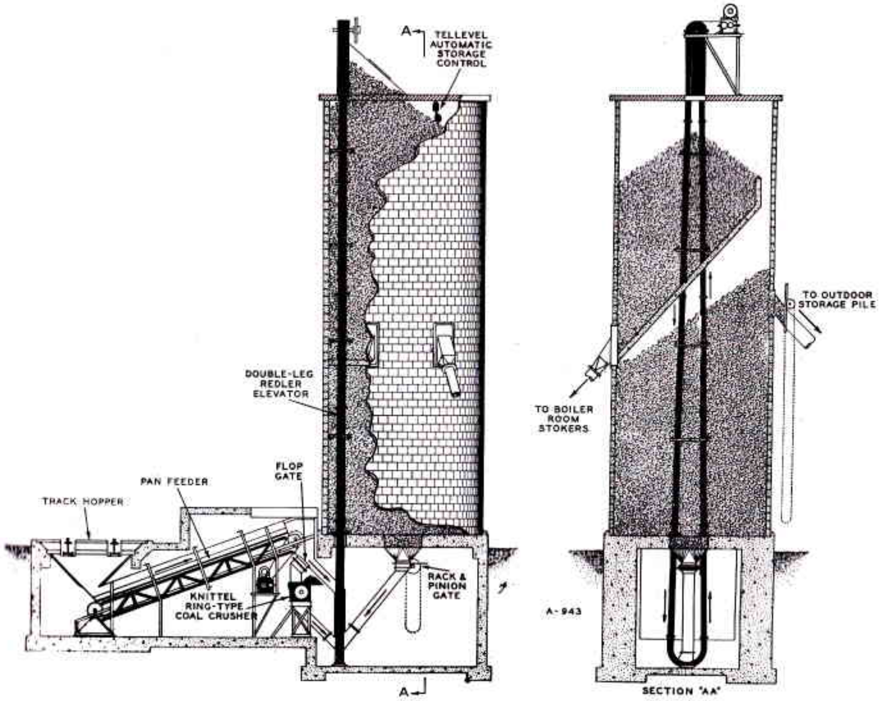


FIG. 10-4. Fuel-handling equipment for a small-capacity plant. Note relation to flow chart (Fig. 10-1). (*Courtesy of Stephens-Adamson Manufacturing Company.*)

Small plants may use lift trucks to move coal from storage to stoker or pulverizer hopper. Most plants, however, use simple coal chutes from overhead bunkers. The chutes, made of corrosion- and abrasion-resistant materials, flare slightly in the flow direction. Steep inclines aid flow, and mechanical vibrators prevent coal from bridging or clogging in chutes and pipes.

Coal may be weighed at the unloading point and at individual stoker or pulverizer hoppers. Track or platform scales may be used for rail cars and trucks. Weighing devices may be built into rotary car dumpers, in feeders, and in belts for coal in transit. Weigh larries measure coal as taken from bunkers and after discharge to hoppers. Automatic

coal-weighing devices are available, as well as automatic samplers for **heating-value** determinations. Level indicators measure how much **coal bins** and hoppers contain.

10-4. Plant Coal-handling Installations. Figure 10-4 shows an installation for a small plant. Equipment arrangement may be identified as in Fig. 10-1. Rail-delivered coal drops to a track hopper and flows to a feeder that delivers the coal to a crusher or directly to an elevator. The silo has interior live and reserve storage volumes and an overflow chute to outside storage. Figure 10-5 shows the arrangement for handling coal in a medium-sized plant.

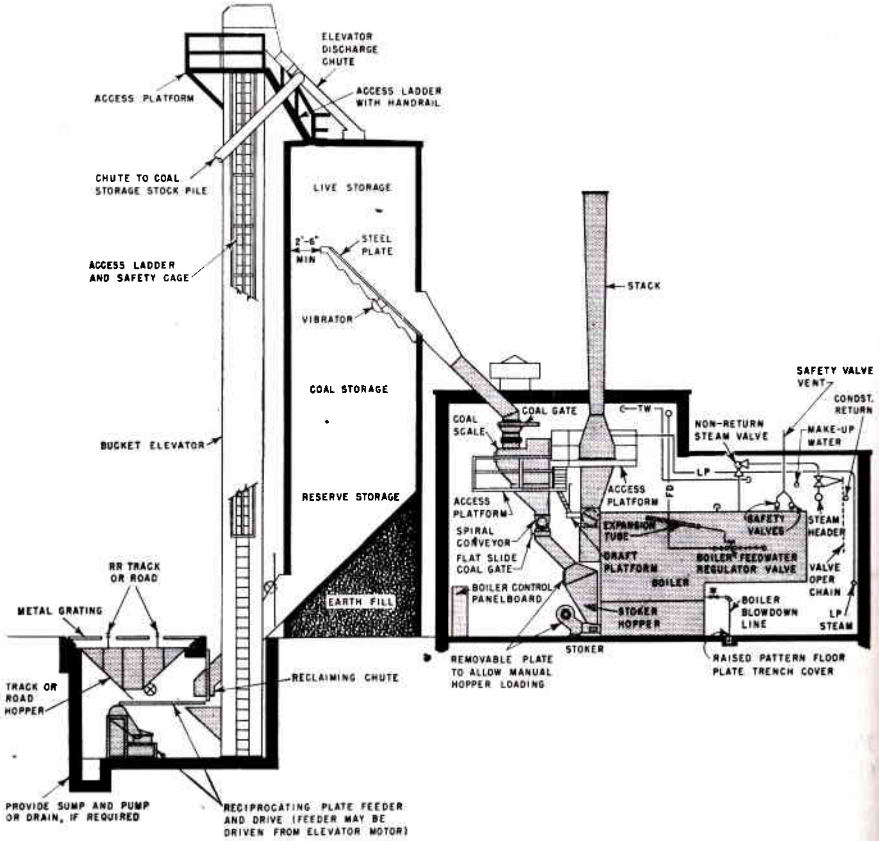


FIG. 10-5. Typical coal-handling layout for medium-sized stoker-fired boiler plant using silo with live and reserve storage. (*Courtesy of Industry Power.*)

Figure 10-6 shows a complete handling system for a large plant that uses thawing pits, track hoppers, bradford breakers, coal dryers, silos, bunkers, weightometers, trippers, and other devices.

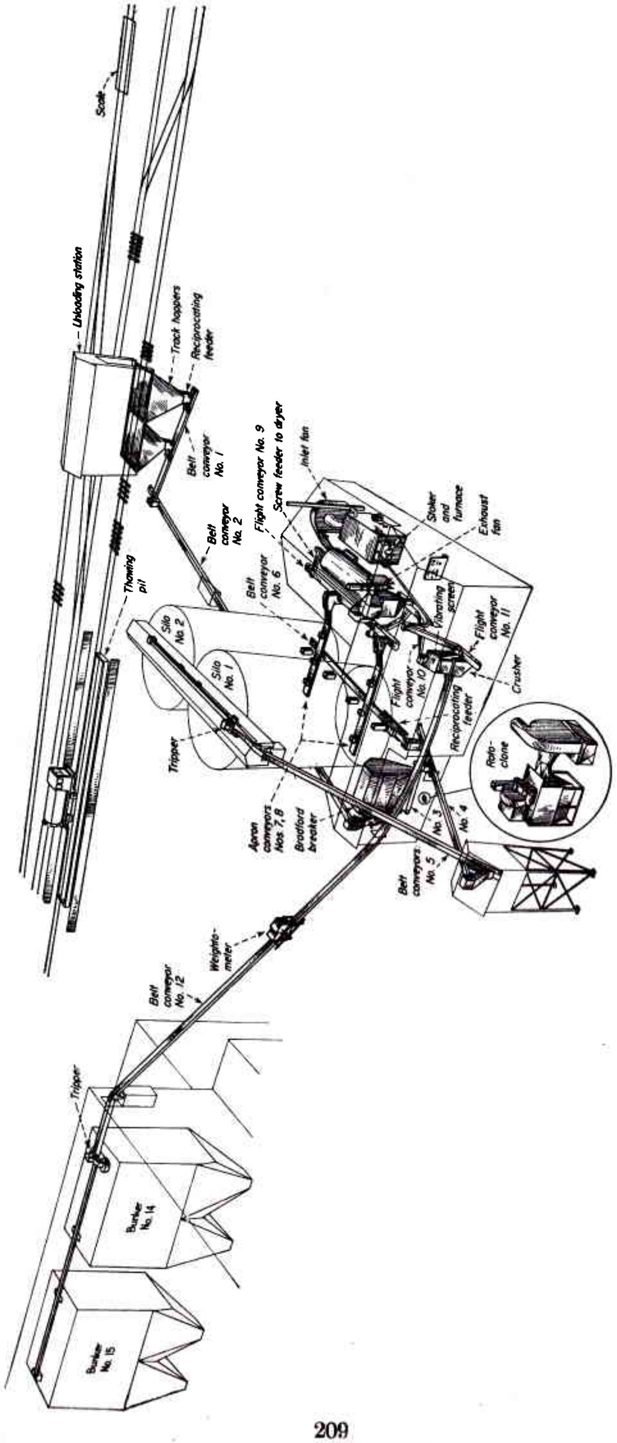
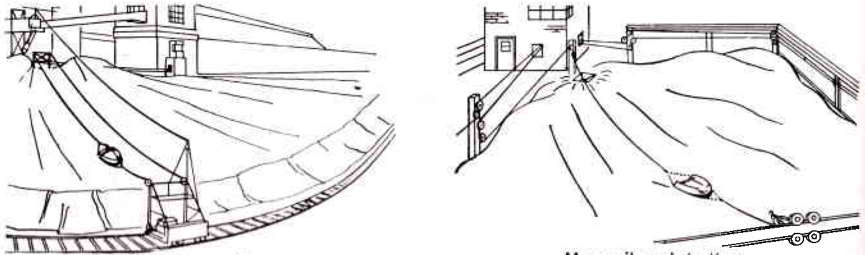


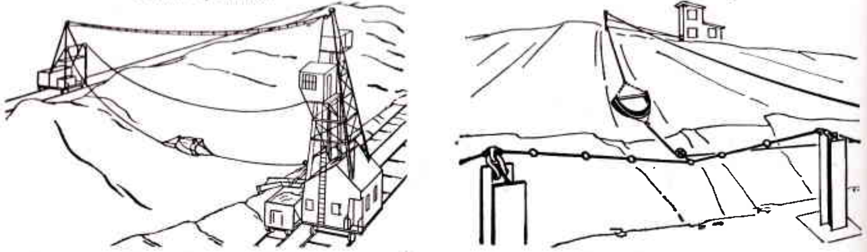
FIG. 10-6. Typical coal-handling system for large coal-fired power plant has coal dryer, bradford breaker, and other features, for drying coal and breaking to certain maximum size.

10-5. Coal Storage. Storing coal guards against interruption of supplies and allows purchasing coal when prices are favorable. Small plants usually store coal under cover in silos (Figs. 10-4 to 10-6). These have live storage for immediate use and reserve storage for future use. Coal goes from silo to stoker hopper by chute or weigh larry. Larger plants use bunkers over or next to firing areas. Exhaust fans keep covered bunkers and silos under a slight negative pressure to minimize dust nuisance.

Coal reserves for large plants are usually stored outdoors next to the plant. Stock-piling methods may use fixed or mobile equipment. Figure 10-7 shows the drag scraper as one of the fixed types; locomotive cranes and bridge cranes with limited range on rails are other types.


Radial operation

Monorail and trolley


Parallel operation

Back posts and bridle

Fig. 10-7. Typical drag scraper layouts for outdoor storage piles. (*Courtesy of Sauerman Bros., Inc.*)

Figure 10-8 shows a carryall for stocking and reclaiming coal from storage. Carryalls are used in large storage piles as in Fig. 10-9. They pick up coal from the stacker-conveyer discharges to store in the pile. On reclaiming they pick up coal from the pile and drop it into the nearest yard hopper.

Stored coal weathers and may ignite spontaneously. Weathering releases some of the volatile matter, to reduce heating values as much as 3 per cent during the first year. It also reduces the ignition quality and reduces the coal size. Slow oxidation of the coal may under certain conditions heat the coal to the point of ignition. Ventilating the pile can sometimes remove the heat and avoid burning. More often coal piles are built up in 1-ft layers and thoroughly compacted by running the tractors and carryalls over them. This excludes air to a large degree and reduces oxidation and so spontaneous heating. For long storage periods sealing the outside surface with asphalt or fine coal makes the pile practically airtight.

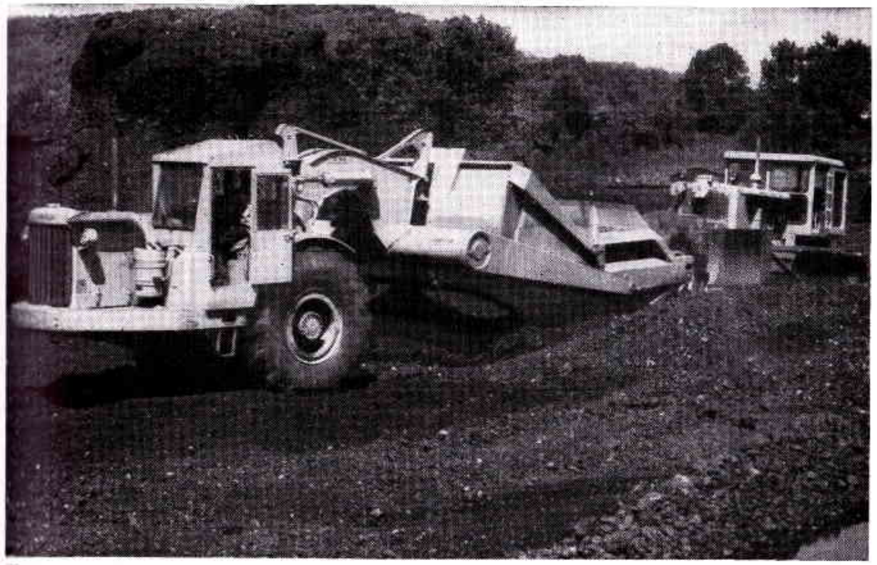


Fig. 10-8. Carryall, left, distributes coal to any place in storage area; it can also pick up coal and deliver it to yard hopper for plant use. Bulldozer levels and compacts piles and acts as pusher when carryall picks up load of coal through bottom opening. (*Courtesy of Caterpillar Tractor Company.*)

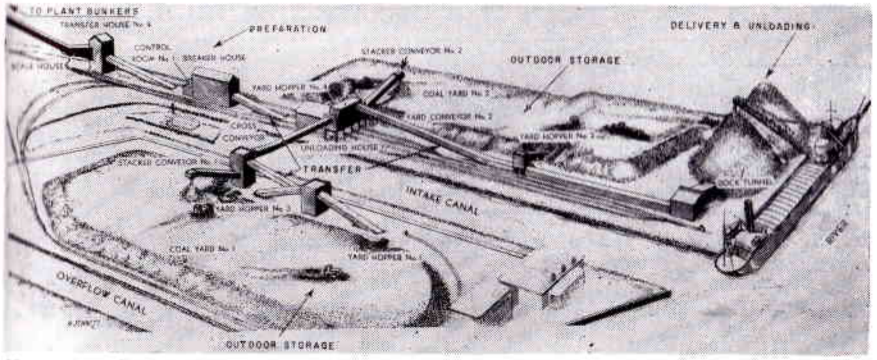


Fig. 10-9. Conveyer layout in storage area of large power plant provides for stockpiling coal delivered by ships. Mobile equipment distributes coal to piles and reclaims coal for use in plant by delivering it to yard hoppers. Breaker house crushes coal. (*Courtesy of Link-Belt Company.*)

10-6. Water Removal. A few plants burn coal fines dredged from river bottoms. These coal fines accumulate from wash water used by upstream mines in their coal-cleaning process. River coals have high moisture content and must be dried before storing or firing. Dryers of various types, screen, rotary kiln, or flash, use flue gas or other heat sources for drying coal (Fig. 10-6).

10-7. Ash-handling Systems. Industrial and central-station ash-removal systems may be either *hydraulic* or *pneumatic* types. In hydraulic systems the ashes and slag leaving a furnace quench in an ashpit or hopper. At intervals high-velocity water jets discharge the refuse to sluiceways that carry it to a disposal point. This may be a fill area or a storage sump. An ash-handling pump empties the ash to a barge or truck or rail car at intervals (Fig. 10-10).

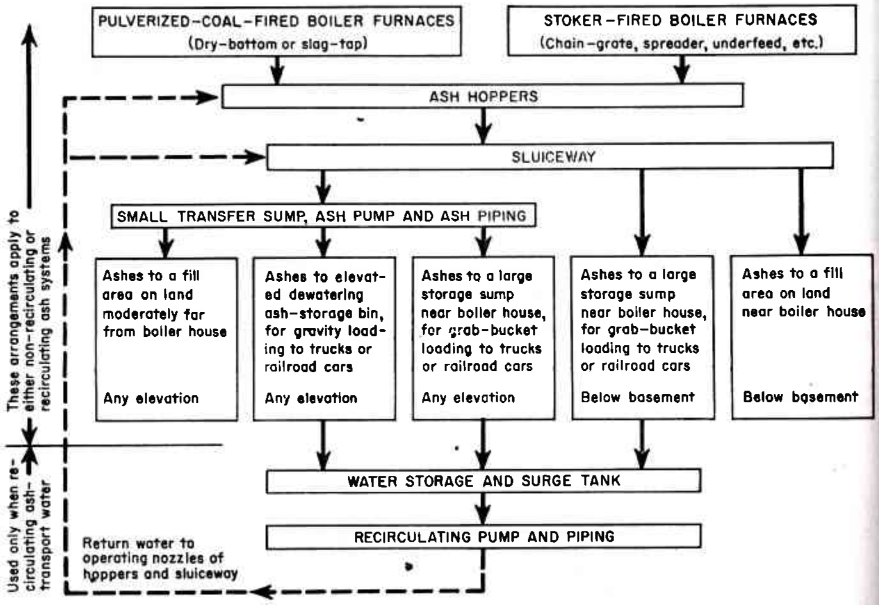

PULVERIZED-COAL-FIRED BOILER FURNACES (Dry-bottom or slag-tap)
STOKER-FIRED BOILER FURNACES (Chain-grate, spreader, underfeed, etc.)
ASH HOPPERS
SLUICEWAY
SMALL TRANSFER SUMP, ASH PUMP AND ASH PIPING
Ashes to a fill area on land moderately far from boiler house
Any elevation
Ashes to elevated dewatering ash-storage bin, for gravity loading to trucks or railroad cars
Any elevation
Ashes to a large storage sump near boiler house, for grab-bucket loading to trucks or railroad cars
Any elevation
Ashes to a large storage sump near boiler house, for grab-bucket loading to trucks or railroad cars
Below basement
Ashes to a fill area on land near boiler house
Below basement
WATER STORAGE AND SURGE TANK
RECIRCULATING PUMP AND PIPING
Return water to operating nozzles of hoppers and sluiceway
These arrangements apply to either non-recirculating or recirculating ash systems
Used only when recirculating ash-transport water

FIG. 10-10. Hydraulic ash-handling system may be applied to a wide variety of conditions. (*Courtesy of Power.*)

In pneumatic systems the ash from furnaces and fly ash from dust collectors and other accumulating points drop into storage hoppers. A high-velocity air stream picks up the ash to carry it through a conveying line to a cyclone separator or target box in an ash-storage bin (Fig. 10-11). The air stream induced by a fan or ejector exhausts to atmosphere. Pneumatic systems handle fly ash effectively because of its fine size, 75 per cent or more through 325-mesh sieve. Some power plants use hydraulic systems for furnace ash and pneumatic systems for fly ash.

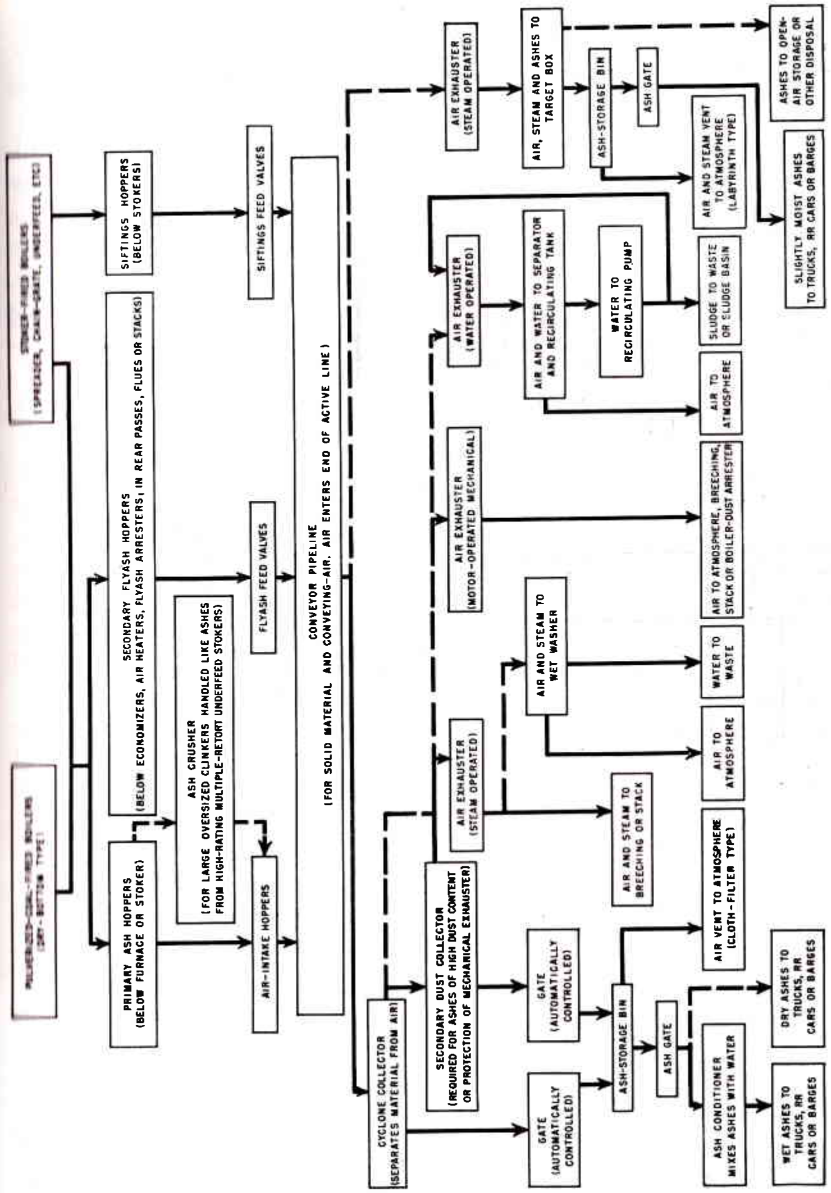

PULVERIZED-COAL-FIRED BOILERS (DRY-BOTTOM TYPE)
STOKER-FIRED BOILERS (SPREADER, CHAIN-GRATE, UNDERFEED, ETC)
SIFTINGS HOPPERS (BELOW STOKERS)
SECONDARY FLYASH HOPPERS (BELOW ECONOMIZERS, AIR HEATERS, FLYASH ARRESTERS, IN REAR PASSES, FLUES OR STACKS)
PRIMARY ASH HOPPERS (BELOW FURNACE OR STOKER)
ASH CRUSHER (FOR LARGE OVERSIZED CLINKERS HANDLED LIKE ASHES FROM HIGH-RATING MULTIPLE-RETORT UNDERFEED STOKERS)
SIFTINGS FEED VALVES
FLYASH FEED VALVES
AIR-INTAKE HOPPERS
CONVEYOR PIPELINE (FOR SOLID MATERIAL AND CONVEYING-AIR. AIR ENTERS END OF ACTIVE LINE)
CYCLONE COLLECTOR (SEPARATES MATERIAL FROM AIR)
SECONDARY DUST COLLECTOR (REQUIRED FOR ASHES OF HIGH DUST CONTENT OR PROTECTION OF MECHANICAL EXHAUSTER)
AIR EXHAUSTER (STEAM OPERATED)
AIR EXHAUSTER (WATER OPERATED)
AIR EXHAUSTER (MOTOR-OPERATED MECHANICAL)
AIR EXHAUSTER (STEAM OPERATED)
AIR, STEAM AND ASHES TO TARGET BOX
ASH-STORAGE BIN
ASH GATE
ASHES TO OPEN-AIR STORAGE OR OTHER DISPOSAL
AIR AND STEAM VENT TO ATMOSPHERE (LABYRINTH TYPE)
SLIGHTLY MOIST ASHES TO TRUCKS, RR CARS OR BARGES
AIR AND WATER TO SEPARATOR AND RECIRCULATING TANK
WATER TO RECIRCULATING PUMP
SLUDGE TO WASTE OR SLUDGE BASIN
AIR TO ATMOSPHERE
AIR TO ATMOSPHERE, BREECHING, STACK OR BOILER-DUST ARRESTER
AIR AND STEAM TO WET WASHER
WATER TO WASTE
AIR TO ATMOSPHERE
AIR AND STEAM TO BREECHING OR STACK
GATE (AUTOMATICALLY CONTROLLED)
GATE (AUTOMATICALLY CONTROLLED)
ASH-STORAGE BIN
ASH GATE
AIR VENT TO ATMOSPHERE (CLOTH-FILTER TYPE)
ASH CONDITIONER MIXES ASHES WITH WATER
DRY ASHES TO TRUCKS, RR CARS OR BARGES
WET ASHES TO TRUCKS, RR CARS OR BARGES

FIG. 10-11. Pneumatic ash-handling systems have particular advantages for handling fly ash. (*Courtesy of Power.*)

10-8. Fuel-oil Handling. Oil-burning power plants usually use Bunker C or No. 6 heavy oil. The National Board of Fire Underwriters Code and local ordinances specify the basis for oil-system designs (Fig. 6-13).

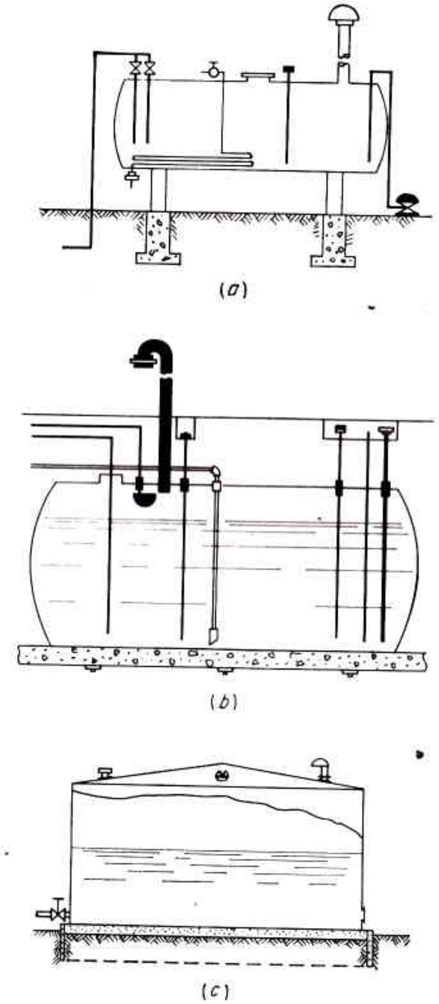


Fig. 10-12. Fuel-oil storage-tank arrangements: (a) outdoors aboveground; (b) underground; (c) large cone-roofed tank. (*Courtesy of Power.*)

Oil delivered to a storage tank flows through a strainer on the way to the oil-pump suction. The pump discharges oil through heaters and another strainer to the burners. Straining, pumping, and heating equipment in duplicate allow cleaning while maintaining oil flow.

Oil arrives at a plant by ship, barge, tank car, or truck and is usually stored close to the unloading point. Storage tanks usually are fitted with fill pipe, vent pipe, oil gauge, steam smothering line, sludge pumpout, manhole, low- and high-suction taps, suction heating coil, return line, and electrical ground.

The steel tanks may be inside or outside the plant. Cylindrical tanks may be installed above ground or below ground or as semiburied tanks (Fig. 10-12).

10-9. Fuel-gas Handling. Pipeline systems make the natural gas found in some 30 states available almost nationwide. For best economy these pipelines should be kept flowing at full capacity. Many power plants use natural gas on a cutoff basis: the plant buys the gas at a low rate but agrees to stop taking fuel when the pipelines run at peak flows for other customers. These plants then burn oil or coal as an alternate fuel.

Plants cannot justify storing gas on their premises, but underground storage proves feasible in some areas by using depleted gas and oil fields. Natural gas usually needs conditioning after withdrawal from the ground in the form of removing dirt, water, hydrogen sulfide, liquid hydrocarbons, and inert gases; an odorant is often added. Long-distance lines use pressures up to 1,000 psig and distribution pressures of about 200 psig or less.

A typical handling system at a plant taps off from a distribution line to a gas-metering house. The line divides into two parallel lines, each with meters, regulators, and stop valves. The lines reunite into a single line from the metering house to the plant boiler room. In the boiler room, branch lines feed gas through meters and regulating valves to the gas burners in the furnaces.

PROBLEMS

10-1. How is the plant fuel rate related to the coal-handling-system rate?

10-2. What type of operation do the following sequences indicate? Refer to Fig. 10-1.

A-B-E-D-F-G-H-I
A-B-D-C-D-F-G-H-I
A-B-D-C-D-E-D-F-G-H-I

10-3. What function does a feeder serve at an unloading point?

10-4. Describe some methods of handling wet or frozen coal.

10-5. What are some problems in designing a flight conveyer?

10-6. How does a belt tripper operate?

10-7. What are the elements of a skip hoist?

10-8. What design features can reduce power requirements in a skip hoist?

10-9. What are reasons for outdoor storage of coal?

10-10. What effects does outdoor storage have on coal properties?

10-11. Sketch and describe the elements of a hydraulic ash-handling system for a slag-tap furnace discharging an elevated bin emptied by trucks.

10-12. Discuss Fig. 10-11 in terms of the different types of air exhausters shown.

10-13. At what points is fly ash deposited in a steam generator?

10-14. How does a cyclone collector operate?

10-15. What is the purpose of the various components of a fuel-oil tank?

10-16. Sketch a simple piping layout for a plant using natural gas.

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CHAPTER 11

DUST COLLECTION

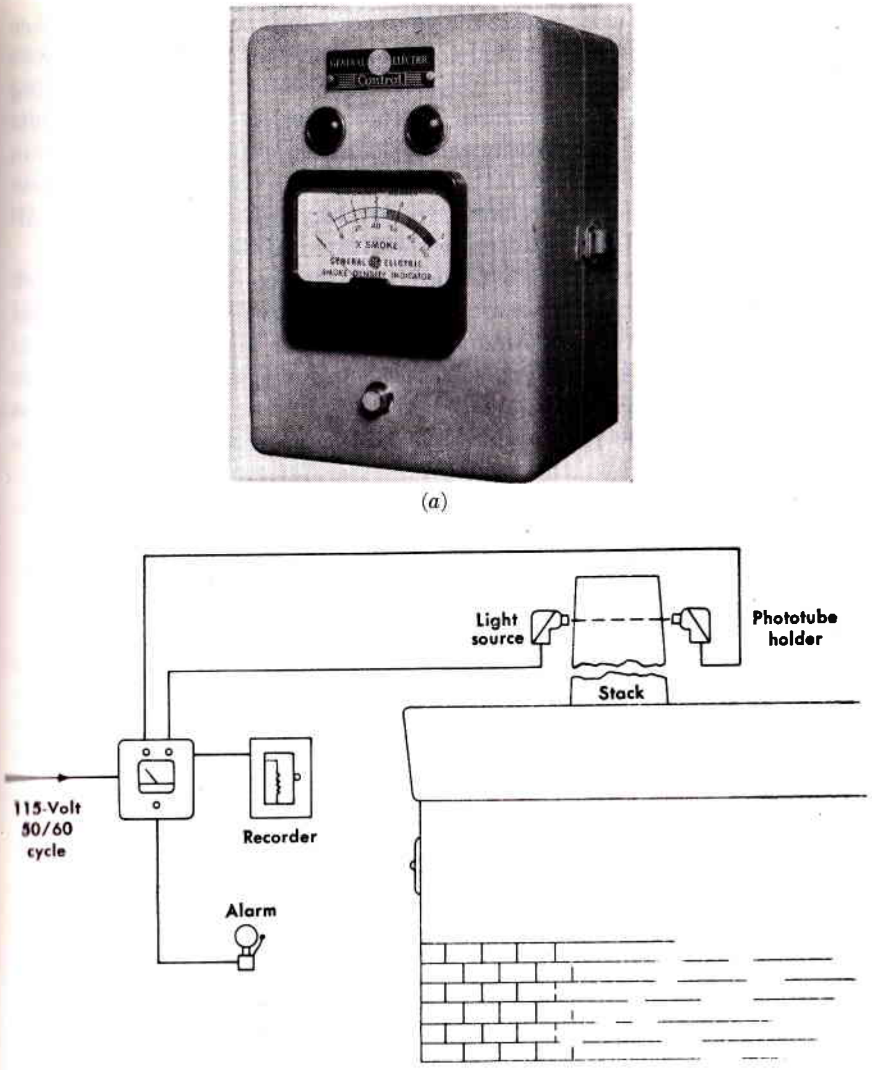
11-1. Dust Types and Measurement. Any gas-borne matter larger than 1 μ (0.001 mm) in diameter we class as dust. When such matter exceeds 100 μ in stack discharges, it is called *cinders;* in smaller sizes it is called *fly ash.* Again, fly ash may refer to gas-borne noncombustible particles and cinders to charred coal and coke particles. Incomplete or arrested combustion of volatile components of fuels produces *smoke,* consisting of particles smaller than 10 μ. When these agglomerate, they form *soot.* Table 11-1 correlates various size measurements.

TABLE 11-1. DUST-SIZE MEASUREMENT UNITS

μ	In.	U.S. sieve series, mesh openings per in.
10	0.00039	Subsieve
20	0.00079	Subsieve
30	0.00118	Subsieve
44	0.0017	325
74	0.0029	200
149	0.0059	100
250	0.0098	60

The dust concentration in stacks and ducts may be found experimentally. The volume, temperature, and pressure of the gas must be measured as well as the weight of the dust. The dust count may be expressed in grains per cubic foot at a specified temperature and pressure. The ASME Power Test Code on Dust Separating Apparatus gives the testing procedure. The size distribution of dust is also analyzed; for instance, a sample may show 5 per cent on 100-mesh, 15 per cent through 100 on 200, 20 per cent through 200 on 325, and 60 per cent through 325.

The Ringelmann chart gives a visual standard for estimating smoke density. It has a series of black crosslines of definite thickness, forming a grid on a white background. Lines of different thickness make up a series of cards numbered 0, 1, 2, 3, 4, and 5. Holding a card at a standard distance from his eyes, an observer can compare it with the smoke in question to estimate its probable density. Many consider the card system outmoded, but it is simple, inexpensive, reproducible, and firmly entrenched in many legal codes.



(a)

(b)

FIG. 11-1. Smoke-density indication. (*a*) Indicator; (*b*) installation in base of stack. (*Courtesy of General Electric Company.*)

Figure 11-1 shows a smoke-density-indicating and -recording system. Such systems use a phototube or thermocouple that measures the intensity of a light beam projected through dust-laden gas entering a stack.

An electronic circuit can operate relays to adjust air supply to the furnace or operate an indicator calibrated to an arbitrary smoke-density scale.

11-2. Air-pollution Control. Aerial contaminants or sewage consisting of dust, smoke, and noxious gases (sulfur oxides) from furnaces affects the physical, economic, and social welfare of a community. The pollution intensity depends on the amount and nature of the contaminants, atmospheric conditions, and the regional topography. Increasing industrialization has increased air pollution in the United States.

Central stations attempt to reduce the solid content of their stack discharges to a minimum in an effort to comply with local ordinances. Legal aspects involve the common law of nuisance and legislative enactments. The latter in the form of codes, ordinances, and air-pollution rules and regulations first appeared in 1867 in St. Louis, Mo. Most cities of over 100,000 population have municipal regulations now. No two communities have the same problem, and so ordinances vary. Legislation usually limits power-plant dust emission but allows the short-duration smoking caused by cleaning fires.

The following is from the 1957 Rules and Regulations of the Department of Air Pollution Control of the City of New York:

1.2.1. No person shall cause, suffer or allow to be emitted into the open air from the stack of any boiler plant or to pass a convenient measuring point adjacent to the stack outlet, within the City of New York or upon waters within the jurisdiction of said city, dust in the gases to exceed the concentrations set forth in the following table except as noted in paragraph (*b*) of this rule:

Steam generated, lb per hr	*Lb dust per* 1,000 *lb steam*
100,000 or less	2.2
150,000	1.9
200,000	1.7
300,000	1.4
400,000	1.2
500,000	1.1
600,000	1.0
800,000	0.9
1,000,000 or more	0.8

Note—For values of steam generated between those tabulated, the corresponding values of pounds dust per 1000 pounds steam may be found by linear interpolation.

(*a*) Steam generated in the above table shall be the average hourly output of all units connected to the chimney for that continuous four hour period during which the average output is the greatest, expressed as the equivalent steam containing 1,000 British thermal units per lb.

(*b*) Dust emission concentration figures shown in the above table may be exceeded during the cleaning periods provided the methods used and the period of time are approved by the Commissioner.

11-3. Atmospheric Dust Dispersion. Fly ash, cinders, obnoxious gases, and smoke discharged from a stack become atmospheric contaminants. Larger particles above 10 μ tend to settle to the ground, while those about 1 μ behave like aerosols, with little tendency to settle. Gases diffuse in all directions. Stack design largely controls the nuisance aspects of aerial wastes. The variables affecting the area over which solids settle out will determine the optimum design of a stack.

The path followed by the gas, called the "plume," depends on the thermal and dynamic properties of the stack gas and the wind flow past the stack. The significant variables are (1) stack height, (2) stack exit-gas velocity, (3) wind velocity, (4) gas temperature, (5) particle size, and (6) surrounding topography. The potential pollution factors of proposed designs can best be determined by wind tunnel test.

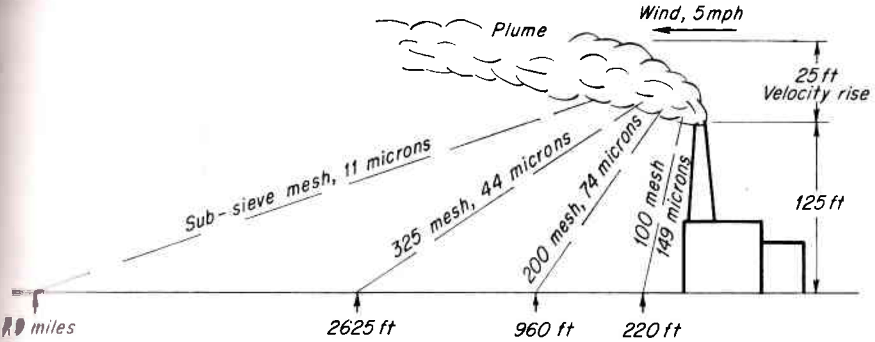


Fig. 11-2. Graphic representation of Stokes's law for a given set of conditions. Dust-settling distances change in proportion to wind velocity.

Stokes's law helps to compute the distance the usual dust sizes will travel from power-plant stacks. Figure 11-2 shows calculation results for a given set of conditions.

The combustible content of stack dust is important in pollution. A typical test showed these combustible distributions: 61.2 per cent on 100-mesh, 43.4 per cent on 200 through 100, 34.7 per cent on 325 through 200, and 15.4 per cent through 325. Coarser particles settle closer to the plant, the black-colored combustible aggravating the nuisance. The average person associates these with the plant, but not the finer, more widely dispersed dust.

11-4. Emissions from Fuel-burning Equipment. Flue-dust quantity and quality depend on the fuel and burning equipment. With pulverized and spreader firing, small particles burn in suspension. They first soften and swell, when heated, to form cenospheres. They form before most of the volatile matter is released. For too short burning times the gas stream carries them out of the furnace.

All types of burning equipment produce flue dust (Fig. 11-3). To collect this dust, the designer must know the dust-size range. The logarithmic probability chart (Fig. 11-4) based on many tests relates particle size and quantity to burning equipment. Variations in fineness of grind, coal composition, and boiler design may shift the curves or change their slope. Table 11-2 lists the dust concentration for different firing equipments.

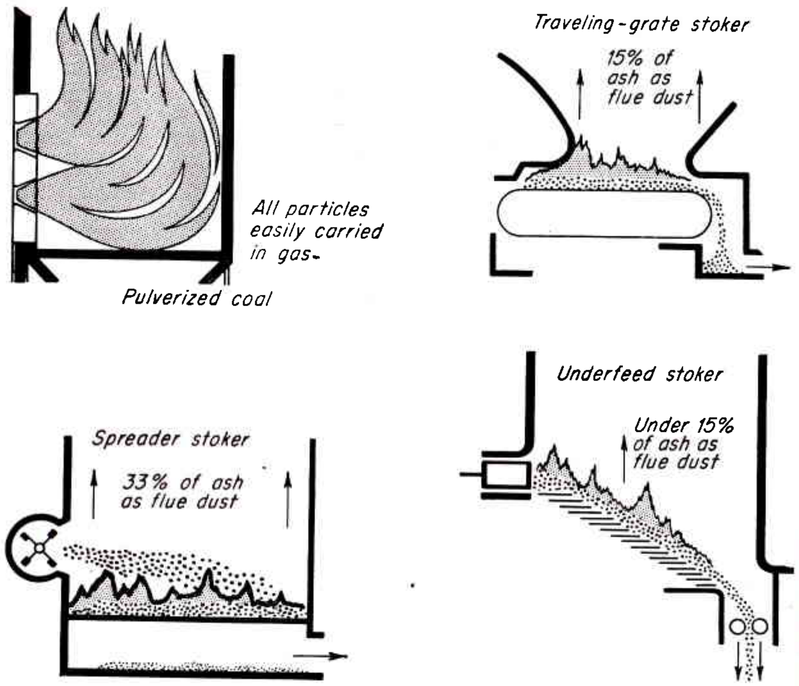


FIG. 11-3. Firing methods influence size, shape, and density of particles and the air-stream velocity with which they leave the furnace and the stack. Flue-dust figures are very approximate. (*Courtesy of Power.*)

TABLE 11-2. TYPICAL AVERAGE DUST CONCENTRATIONS

Equipment	*Dust, grains per cu ft gas*
Pulverized coal	1–12
Chain-grate stoker	2–5
Underfeed stoker	0.5–1.5
Spreader stoker	1–7

11-5. Flue-dust Collection. Figure 11-5 shows the basic principles of mechanical dust collectors. Enlarging the duct cross-sectional area to slow down the gas gives the heavier particles a chance to settle out (Fig. 11-5*a*). When the gas makes a sharp change in flow direction, the heavier particles tend to keep going in the original direction and so settle out (Fig. 11-5*b*). In Fig. 11-5*c* impingement baffles have more effect on the solid particles than the gas, helping them to settle out.

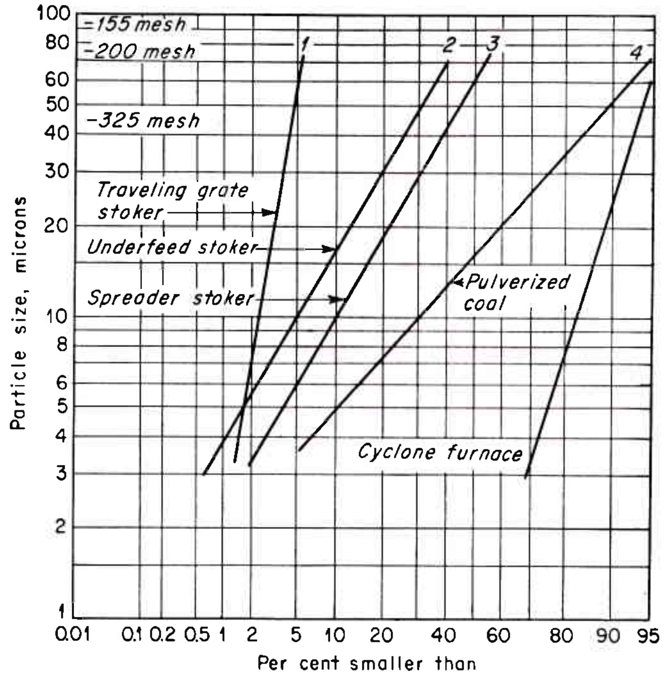


FIG. 11-4. Logarithmic probability chart showing typical distribution of particle size for various types of firing.

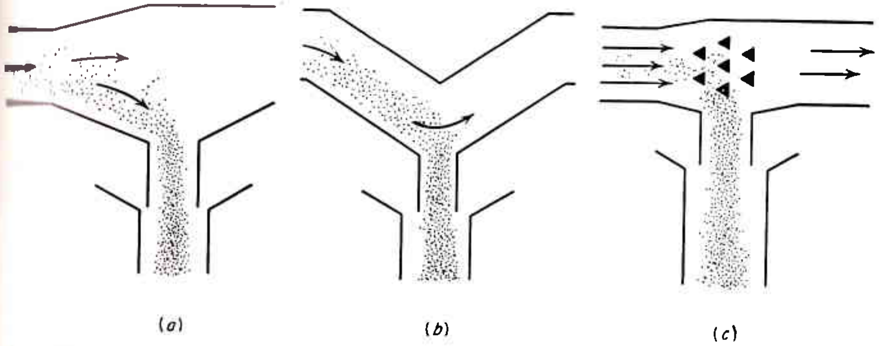
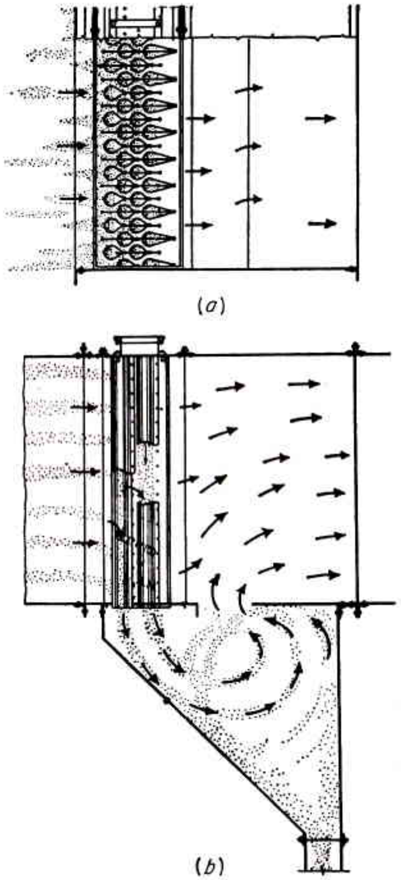


FIG. 11-5. Principles used in dust separation. (*a*) Sudden velocity decrease by enlarging cross section; (*b*) abrupt change of flow direction; (*c*) impingement upon small baffles.

Centrifugal force and forces of electrostatic fields also separate dust from the gas. The *collection efficiency* of a dust separator is the amount of dust removed per unit weight of dust. Though dust collectors remove contaminants, they increase draft losses and hence the fan power.

Let us briefly study some available collector designs.

Cinder Catchers. These are ordinarily used with stoker firing (Fig. 11-6). The plan shows the dust-laden gas striking a series of vertical baffles that change its direction and reduce its velocity. The separated dust and cinders fall to the hopper for removal by a vacuum system. Collection efficiency ranges from 60 to 75 per cent.


(a)

(b)

Fig. 11-6. Typical cinder trap. (*a*) Plan section; (*b*) vertical section.

Cinder Vane Fans. Figure 11-7 shows dust-laden gas from a stoker-fired furnace entering the scroll outer side. While the gas sharply changes direction to enter the rotor, the centrifugal force carries the dust particles along the edge of the scroll. The tangential take-off slot skims a small volume of highly concentrated dust-laden gas to a secondary

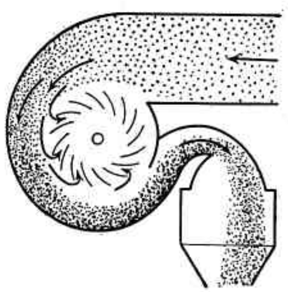


Fig. 11-7. Principle of cinder vane fan, which adds centrifugal force to pull out particles of dust. Note skimmer collecting high-count dust-laden gas.

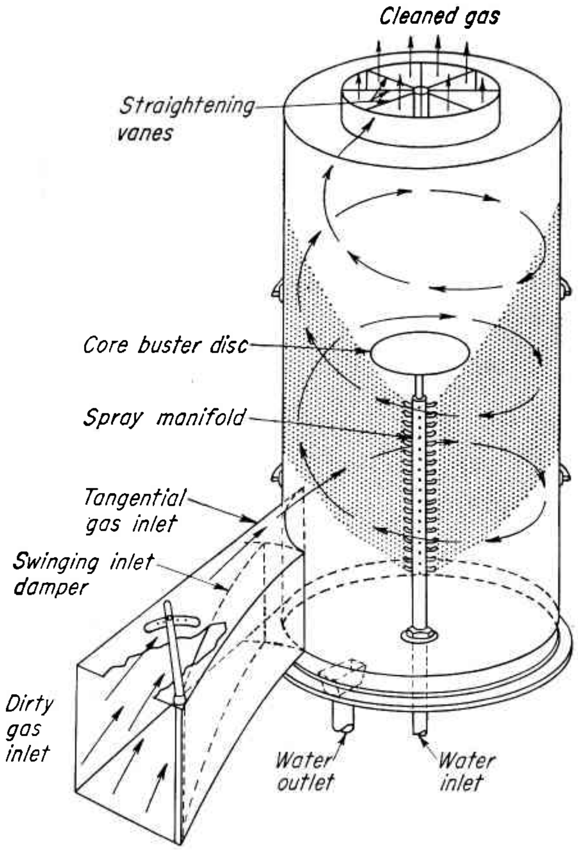


Fig. 11-8. Schematic view showing elements of cyclonic spray scrubber. (*Courtesy of Chemical Construction Corporation.*)

separator. Clean gas at the fan center passes out through special guide vanes. The collecting efficiency varies from 50 to 60 per cent.

Gas Scrubbers. Figure 11-8 shows the wet removal of flue dust and sulfur gases by water-spray washing. A manifold in the center sprays water through the incoming gas, and the dust flows out in a water slurry through the drain below. Rubber-lined pipes, vitrified materials for nozzles, and lead lining in the tanks minimizes corrosion. The collection efficiency runs over 90 per cent, but the corrosion problem precludes wide adoption in power plants.

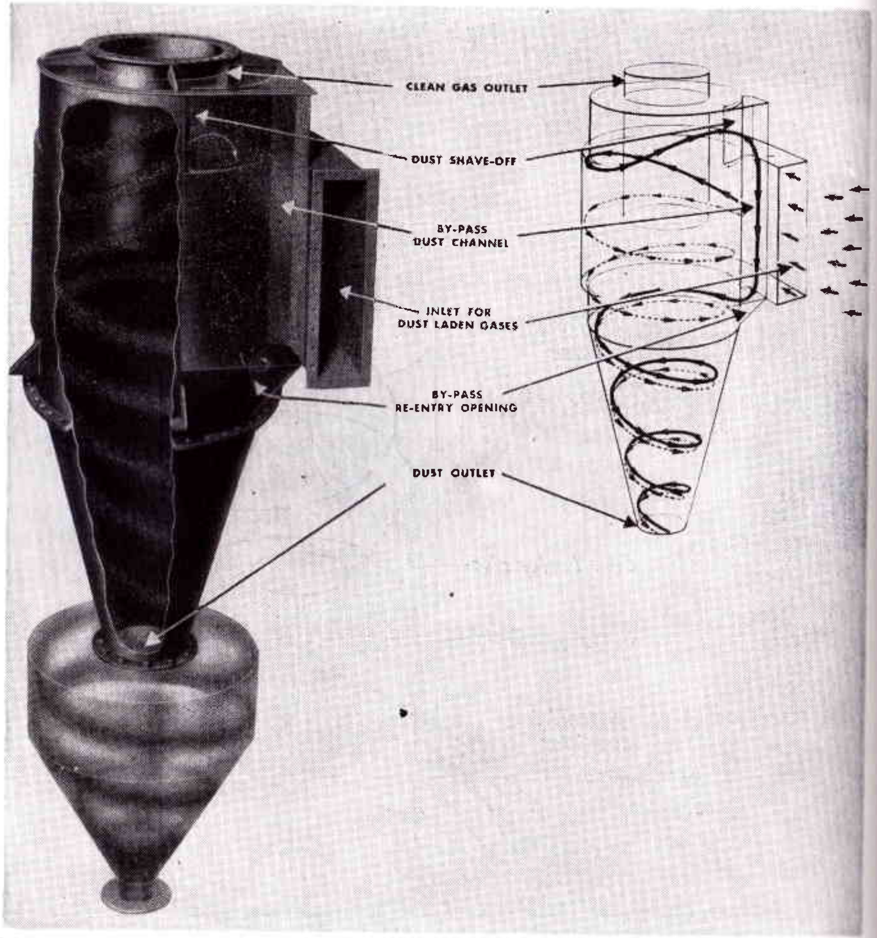


FIG. 11-9. Typical cyclone collector. (*Courtesy of Buell Engineering Company, Inc.*)

Cyclonic Separators. This unit uses a downward-flowing vortex for dust-laden gases along the inner walls (Fig. 11-9). Cleaned air leaves from an inner upward-flowing vortex. An involute or tangential type of

inlet produces vortex flow. Dust particles thrown to the walls by centrifugal force fall to the bottom for removal. This separator has a "shave-off"—a narrow, vertical rectangular port in the upper casing where the upper half of the double eddy current tends to accumulate fines. These enter the shave-off port to flow downward and join the main body of dust below. These cyclones may be used singly or in parallel; they run with efficiencies of about 85 per cent.

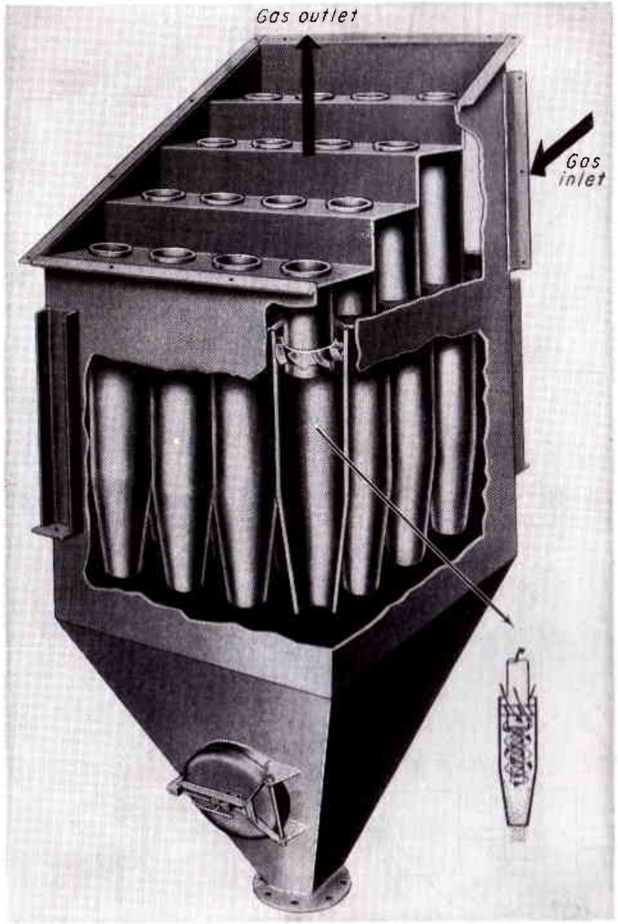


FIG. 11-10. Cyclone separator using a bank of small collecting tubes in parallel. Note vanes for spiraling effect. (*Courtesy of Western Precipitation Company.*)

Figure 11-10 shows a unit using a bank of small-diameter cyclones in parallel. Helical fanlike vanes in the annular space leading to the collecting tubes impart a spiral flow to the gas for centrifugal separation. This type uses a 9-in. tube, while the larger types may have a

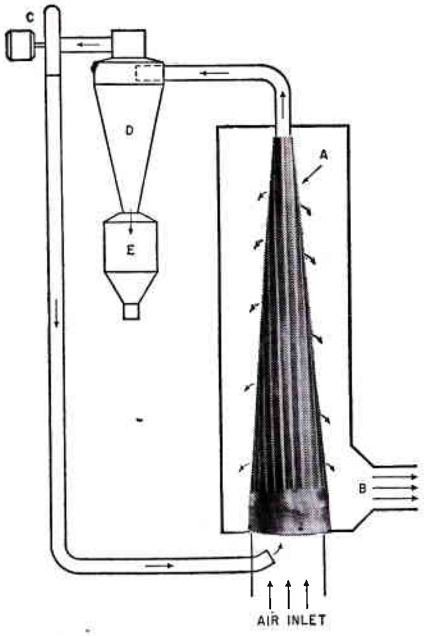


Fig. 11-11. Louver type of separator. Dust-laden air enters at bottom of cone *A*, and clean air escapes through the many fine louvers. Concentrated dust-laden gas is drawn by fan *C* through centrifugal collector *D*. *(Courtesy of The Green Fuel Economizer Company, Inc.)*

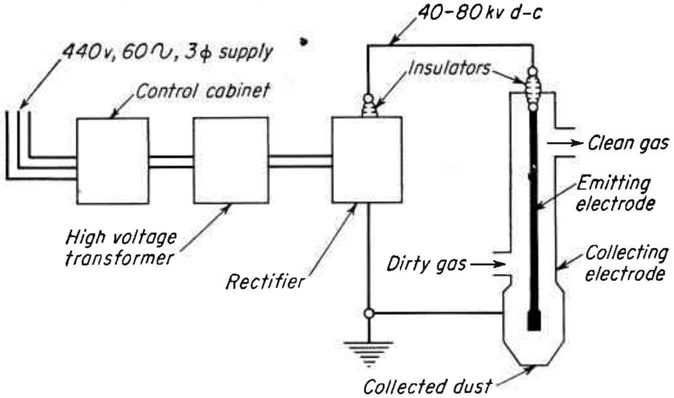


Fig. 11-12. Basic elements of electrostatic precipitator (control cabinet, high-voltage transformer, d-c rectifier, collector).

4-ft diameter. Avoiding high velocities and using wear-resistant materials reduce the erosion in these separators.

Louver-type Separator. Figure 11-11 shows this high-velocity type, which may be classified between the baffle and cyclonic types. A conical shell of concentric overlapping rings guides the dust into the apex of the cone while discharging cleaned gas along the sides. Power plants use these filter cones in parallel.

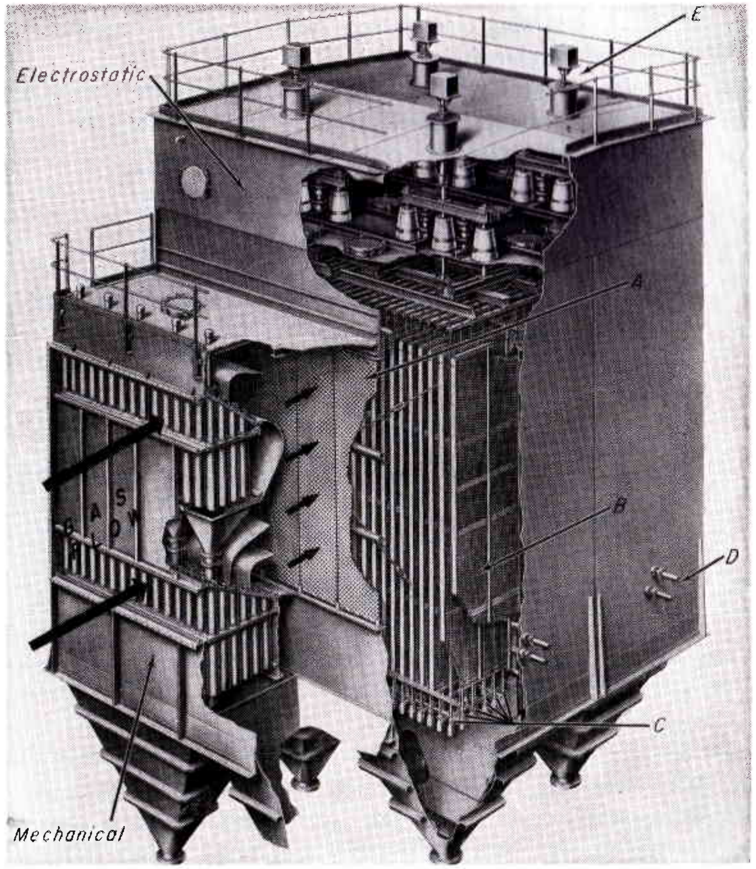


Fig. 11-13. Combination mechanical and electrostatic precipitator. *A*, perforated distribution plates; *B*, grounded collecting plates; *C*, discharge electrodes, high voltage 50 to 70 kv; *D*, plate rappers; *E*, discharge electrode rappers. *(Courtesy of Research-Cottrell, Inc.)*

Electrostatic Precipitators. This type (also called "Cottrell precipitators") work effectively on the finer flue dusts. The precipitator (Fig. 11-12) has two sets of electrodes, insulated from each other, that maintain an electrostatic field between them at high voltage. The field ionizes dust particles that pass through it, attracting them to the electrode of opposite charge. The high-voltage system maintains a negative potential of 30,000 to 60,000 volts with the collecting electrode grounded. The collecting electrodes have a large contact surface. Accumulated dust falls off the electrode when it is rapped mechanically.

Figure 11-12 shows the principal parts: (1) source of high voltage, (2) ionizing and collecting electrodes, (3) dust-removal mechanism, and (4) shell to house the elements.

A wet type of this unit removes dust by a water film flowing down on the inner side of the collecting electrode. These units have collection efficiencies of the order of 90 per cent.

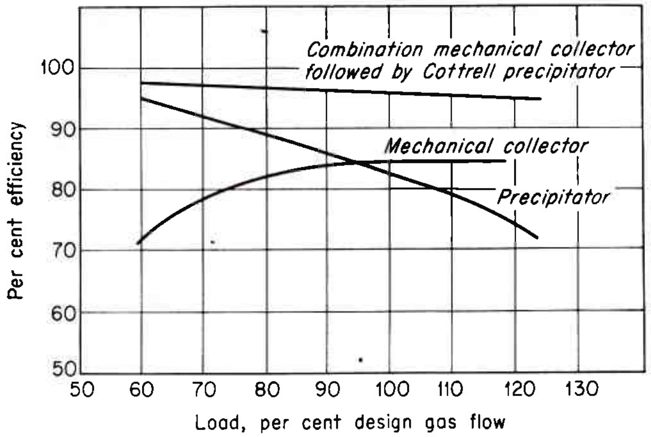


FIG. 11-14. Typical efficiency curves for combination mechanical and Cottrell electrostatic precipitator when used for fly ash.

Figure 11-13 shows a combination mechanical and electrical dust collector. The electrostatic precipitator is costly but has low maintenance needs. The combination unit works well where there is a reasonable percentage of +10-μ particles in the dust and dust loadings of the order of 25 grains per cu ft. The opposing efficiency characteristics of mechanical and electrical separators give a uniform efficiency over the load range (Fig. 11-14).

11-6. Flue-dust Disposal. Flue dust may be used for fill when mixed with heavier components. Commercial uses have been found in some areas that give a return; these include partial replacement of cement in concrete, filler in asphalt for roads, and component in concrete building blocks.

Some slag-tap steam generators use a system of refiring fly ash collected from separators. This turns it into more easily disposable slag. When the fly ash contains some combustible, the heating value is recovered. Nozzles blow the fly ash back into the furnace.

Many English plants treat flue gas to recover sulfur dioxide as a profitable by-product.

PROBLEMS

11-1. How many openings per square inch has a sieve sized for 74-μ particles?

11-2. Which ASME power test code is used in this field? What does it cover?

11-3. Discuss some of the advantages and disadvantages of a Ringelmann chart.

11-4. What maintenance problems might be encountered in smoke-density control systems?

11-5. Briefly discuss some problems that are encountered in setting up a code for air-pollution control.

11-6. Over what factor does a designer have control from the standpoint of dust dispersion from a stack?

11-7. What is the general relation between dust size and combustible content?

11-8. Briefly classify commercial types of dust collectors.

11-9. What are the main elements of an electrostatic precipitator?

11-10. Suggest some possible uses of fly ash collected in precipitators.

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CHAPTER 12

STEAM PRIME MOVERS

12-1. Turbine Development. The history of turbine development starts with the rotating-reaction nozzle-equipped sphere of Hero of Alexandria about 120 B.C. In 1831 Foster and Avery obtained a United States patent for a reaction wheel similar to Hero's. DeLaval first worked with the reaction idea but became famous for his single-stage impulse turbine about 1889. Parsons made a commercial success of the reaction turbine about 1894 in a marine application. Curtis developed the velocity staged turbine about 1896.

Active development of the turbine made it the principal prime mover of generating stations by 1920. Most units used 200-psig 550-F steam, and capacities ranged from 5,000 to 30,000 kw. Small units ran at 3,600 rpm and larger ones at 1,800 rpm. Turbine arrangements included straight condensing, back pressure, and automatic extraction.

By 1930 steam conditions rose to 700 psig and 750 F, with a few units at 1,200 psig. Single-cylinder units of 15,000 kw at 3,600 rpm and 75,000 kw at 1,800 rpm were running. The largest were 160,000-kw compound units running at 1,800 rpm.

By 1940 turbines were using 950-F steam, and pressures above 1,200 psig were accepted. Since World War II the reheat cycle has been widely adopted for new large units.

In 1957 the first supercritical unit with two stages of reheat began commercial service. Steam conditions were set at 4,500 psig and 1150/1050/1000 F; this Philo unit had a rating of 125 mw. Other supercritical units that followed within a few years were built for as high as 450 mw, but pressure was backed down to 3,600 psig to minimize difficulty with first-stage turbine bucket design.

12-2. Basic Turbine Forms. Allowing pressurized steam to flow through a nozzle to a lower pressure converts part of the steam energy to kinetic jet energy. Turbines use two methods: (1) impulse and (2) reaction. Both methods are used in some impulse-reaction turbines.

Figures 12-1 and 12-2 show features of these two main types of turbines. Individual designs vary widely.

Impulse Turbine. The impulse turbine (Fig. 12-1) expands the steam

in stationary turbine nozzles, where it attains a high velocity. The steam jet flows over moving blades without further expansion. These blades or buckets are fastened to the rims of rotating disks mounted on the turbine shaft. This turbine uses Curtis (velocity) and Rateau (pressure) stages.

Reaction Turbine. The reaction turbine (Fig. 12-2) expands the steam in both fixed blades and moving buckets on the rim of a rotating drum. The relative amount of expansion varies with the particular design.

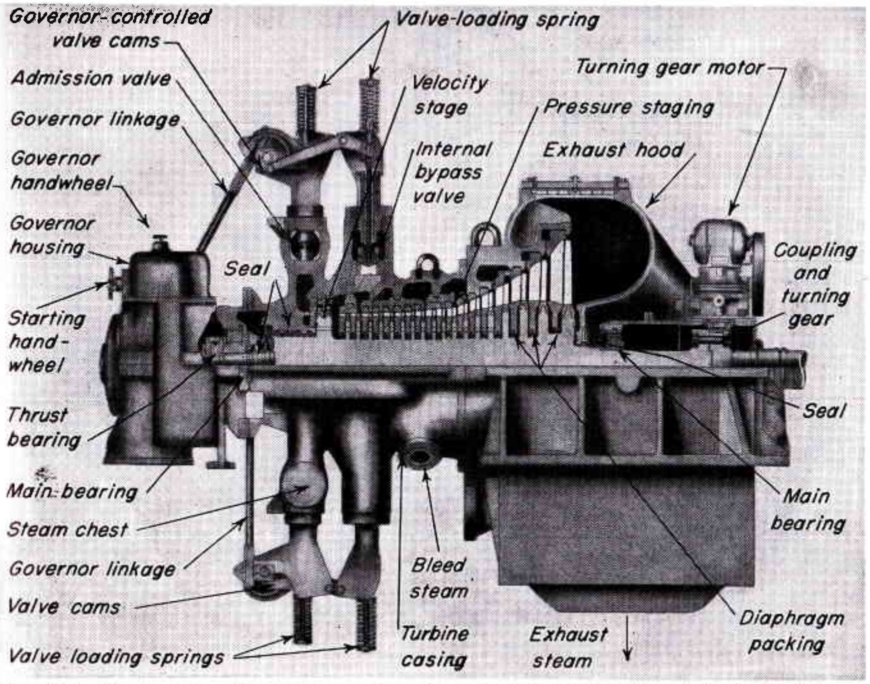


FIG. 12-1. Typical impulse turbine. Impulse principle used in Curtis (velocity) stage at high pressure end followed by a series of Rateau (pressure) stages. Note stationary nozzles and blades attached to disks which rotate shaft. (*Courtesy of General Electric Company.*)

12-3. Turbine Steam-flow Patterns. Figure 12-3 shows the basic layout of turbine nozzles, blades, and buckets. The pressure-staged unit has alternating rows of stationary nozzles and moving buckets. The simplest turbines have a single nozzle and one rotating disk carrying buckets on its rim. Larger units have multiple pressure stages to accommodate larger pressure drops. The moving blades develop the torque to turn the disks that carry them. The nozzles are in blocks at the first or early stages and are placed in stage-dividing members called "diaphragms."

Velocity-staged flow increases the number of rows of moving buckets per stage. The first row of buckets absorbs only part of the jet kinetic energy—a set of stationary blades redirects the steam into a second row of moving buckets without dropping the steam pressure.

The reaction flow shown is typical of 50 per cent reaction turbines. Alternate fixed and moving blades with blunt-nosed entrance edges drop the steam pressure in each step to convert steam energy to jet kinetic energy.

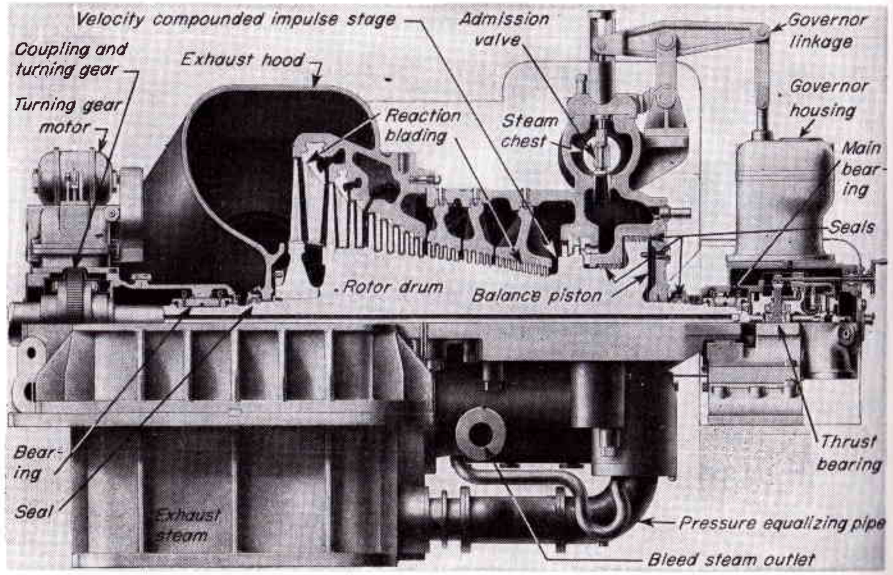


FIG. 12-2. Typical reaction turbine. Note fixed and moving blades formed as nozzles. Use of impulse velocity stage at high-pressure end makes this a combination impulse-reaction unit on an over-all basis.

In all turbine steam flow, as pressure drops the velocity relative to the turbine nozzles increases, and the specific volume of the steam rises. The curves in Fig. 12-3 should be taken as showing general trends only and not as accurate characteristics.

12-4. Turbine Nozzle Action. The acceleration of steam flowing through a nozzle to a lower pressure takes place in: (1) stationary nozzles of impulse turbines in the nozzle block at the stage inlet or formed by blade passages in diaphragms, (2) stationary nozzles (blade-formed passages) of reaction turbines, (3) moving nozzles (blade-formed passages) of reaction turbines.

Figure 12-4 shows these arrangements. The steam moves in steady

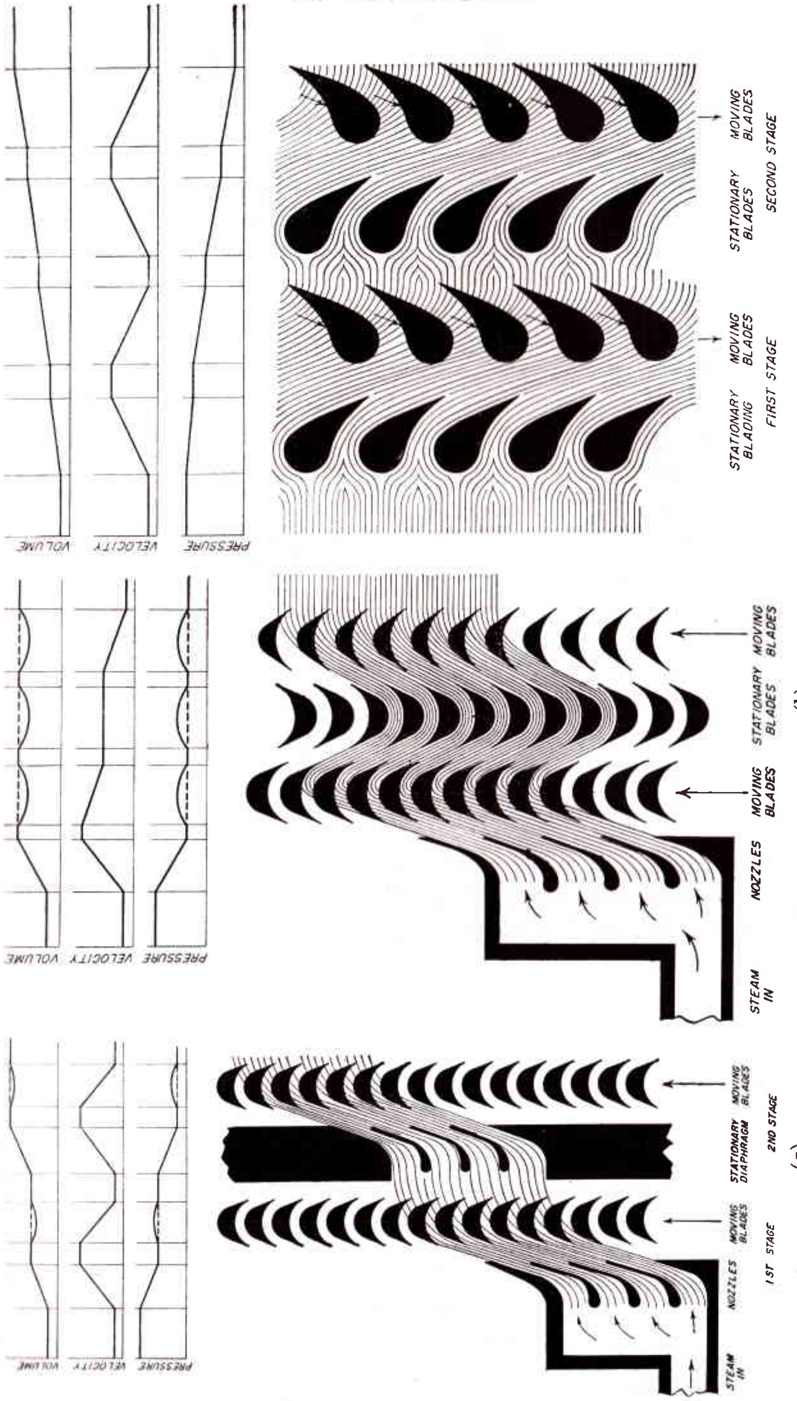


FIG. 12-3. Steam flow. (*a*) Pressure-staged or -compounded (Rateau) impulse turbine; (*b*) velocity-staged or -compounded (Curtis) impulse turbine; (*c*) reaction-turbine blading or nozzles.

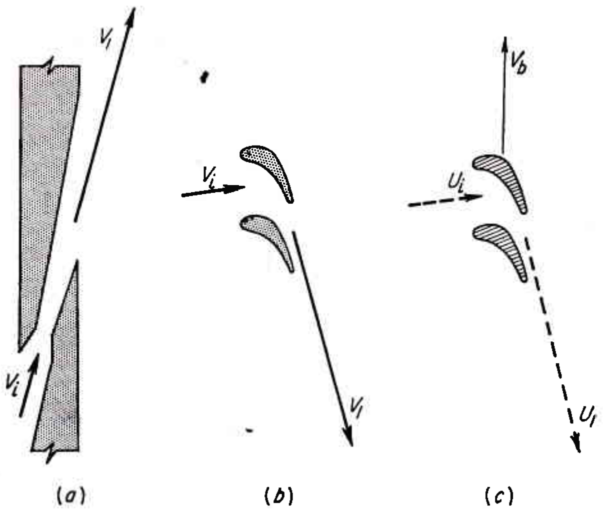


FIG. 12-4. Turbine nozzle action. (a) Impulse inlet; (b) impulse diaphragm and stationary reaction; (c) moving reaction.

flow, and so the steady-flow energy equation may be used for analysis,

$$h_{ti} = h_{t1} \tag{12-1}$$

$$h_{si} + \frac{V_i^2}{2g} = h_{s1} + \frac{V_1^2}{2g} \tag{12-2}$$

$$h_{si} + \frac{U_i^2}{2g} = h_{s1} + \frac{U_1^2}{2g} \tag{12-3}$$

where i, 1 = inlet and outlet conditions
h_s = static enthalpy, Btu per lb
h_t = total, or stagnation, enthalpy
V = absolute velocity, ft per sec
U = relative velocity to moving blades, ft per sec
V_b = moving-blade velocity, ft per sec

We also apply the continuity equation to help the analysis,

$$w = A_iV_i\rho_i = A_1V_1\rho_1 \tag{12-4}$$

$$w = A_iU_i\rho_i = A_1U_1\rho_1 \tag{12-5}$$

where w = mass flow, lb per sec
A = cross-section area of nozzle, sq ft
ρ = fluid density, lb per cu ft

For incompressible fluids the nozzle area decreases from inlet to outlet; for compressible fluids the flow path first converges and then diverges if a large enough pressure drop is used. For isentropic flow the pressure at the minimum, or throat, section is always about 0.55 the inlet pressure for superheated steam and 0.58 for saturated steam; these are absolute pressures. Inlet turbine nozzles may be converging-diverging, but for reaction stages and most latter impulse stages the nozzles are converging only because of the smaller pressure drop per stage.

Figure 12-5 plots the nozzle-expansion line. An ideal nozzle expands isentropically, and all the available energy converts to velocity energy. But turbulence and friction prevent isentropic expansion so that steam enthalpy leaving is higher than the ideal. Equations (12-1) to (12-5) apply also to actual conditions as well as ideal. We calculate performance referred to ideal as

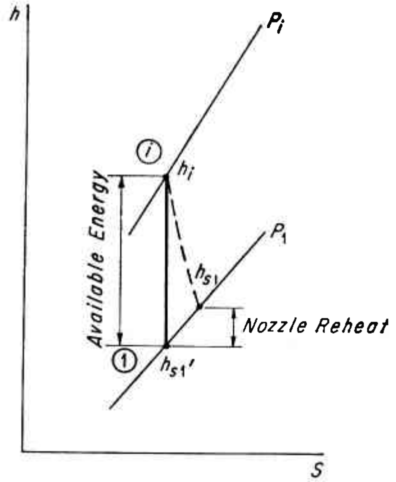


FIG. 12-5. h-s diagram for turbine nozzle action. Note $(h_{s1}\text{-}h_{s1'})$ due to the irreversibility of the actual steam process.

$$(h_i - h_{s1}) = \eta_n(h_i - h_{s1'}) \tag{12-6}$$

where η_n is the nozzle efficiency. The power in the steam jet leaving the nozzle is

$$J = \frac{wV_1^2}{2g} \tag{12-7}$$

where J = jet power in foot-pounds per second.

12-5. Turbine Blade and Stage Action. The moving blade or bucket in a turbine converts the kinetic energy of the steam jet to a torque acting on the turbine shaft. The stationary blade may just change jet direction as in a velocity-compounded stage. Buckets and blades act as: (1) moving blades or buckets in impulse-turbine stages, (2) moving blades in reaction-turbine stages, and (3) reversing blades in velocity stages of impulse turbines.

Figure 12-6 shows the velocity relations about moving blades where

$$P = F \times V_b \tag{12-8}$$

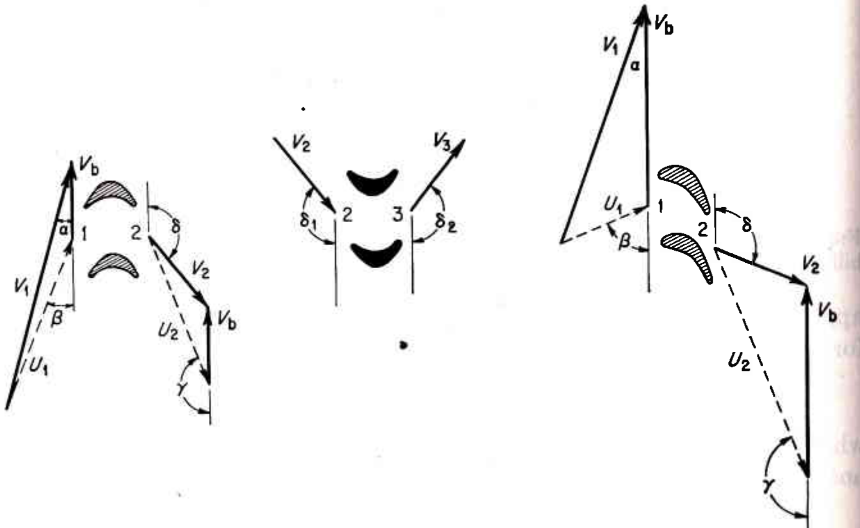
$$P = \frac{w}{g}(V_1 \cos \alpha - V_2 \cos \delta)V_b \tag{12-9}$$

where P = blade power, ft-lb per sec
w = steam flow, lb per sec
F = force exerted by steam on blade = rate of change of tangential momentum, lb
V_b = moving-blade velocity, lb per sec

From an energy balance for a moving blade with adiabatic conditions

$$P = \frac{w}{g}[(V_1^2 - V_2^2) + (U_2^2 - U_1^2)] \tag{12-10}$$

In impulse blades because of friction we find $U_2 < U_1$, whereas in reaction blades even with friction $U_2 > U_1$.



(a) (b) (c)

FIG. 12-6. Turbine blade action. (a) Any moving impulse blade; (b) fixed reversing blades of impulse velocity staging; (c) any moving reaction blade (design shown is for 50 per cent reaction type).

When applying Eq. (12-8) to the stationary reversing blades of a Curtis stage (Fig. 12-6b), we find $P = 0$ since $V_b = 0$. A fixed blade cannot do any work even though the steam exerts a force on it.

Figure 12-7 shows the effect of fluid and blade friction on enthalpy of the steam in moving blades. Frictional losses in Fig. 12-6a make $U_2 < U_1$, in Fig. 12-6b make $V_3 < V_1$, and in Fig. 12-6c show up in U_2

being less than the maximum possible. This frictional energy returns to the steam as increased enthalpy; the energy returned is called *reheat*.

The effectiveness of a blade in converting energy is given by the blade efficiency

$$\eta_b = \frac{P}{J} \tag{12-11}$$

where P = blade power
J = jet power

Impulse-stage blade efficiency can be easily figured by this relation. The expansion in moving blades of reaction stages makes the efficiency more difficult to calculate (see Selected Bibliography).

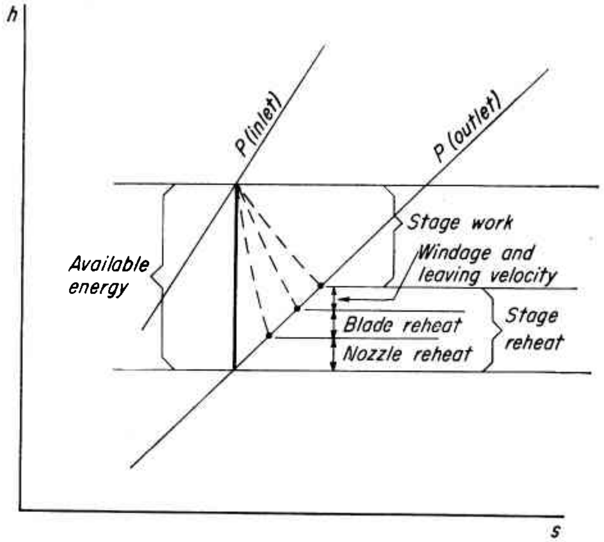


FIG. 12-7. *h-s* diagram for turbine blade action. Note increase of leaving enthalpy due to friction.

The *stage* of an impulse blade includes the nozzles and the blade or bucket rows immediately following it (as many as three rows); the stage of a reaction turbine includes a set of fixed blades and the moving blades following it. Performance is measured by stage efficiency,

$$\eta_s = \frac{W}{(\Delta h)_s} \tag{12-12}$$

where W = blade work per stage, Btu per lb steam
$(\Delta h)_s$ = isentropic drop in enthalpy per stage, Btu per lb

Figure 12-8 shows the physical factors that make the stage of an actual turbine less than 100 per cent efficient. The losses raise the leaving enthalpies above the ideal. These effects repeat in each stage of a multistage turbine. The diverging pressure lines on a Mollier diagram for steam show that with stage reheat $\Sigma[(\Delta h)_s \text{ per stage}] > (\Delta H)_s$ for a single isentropic expansion between inlet and discharge pressure. With this relationship established we can define a reheat factor,

$$R = \frac{\Sigma(\Delta h)_s}{(\Delta H)_s} \tag{12-13}$$

$R > 1$ and helps to estimate the state points for the individual stages.

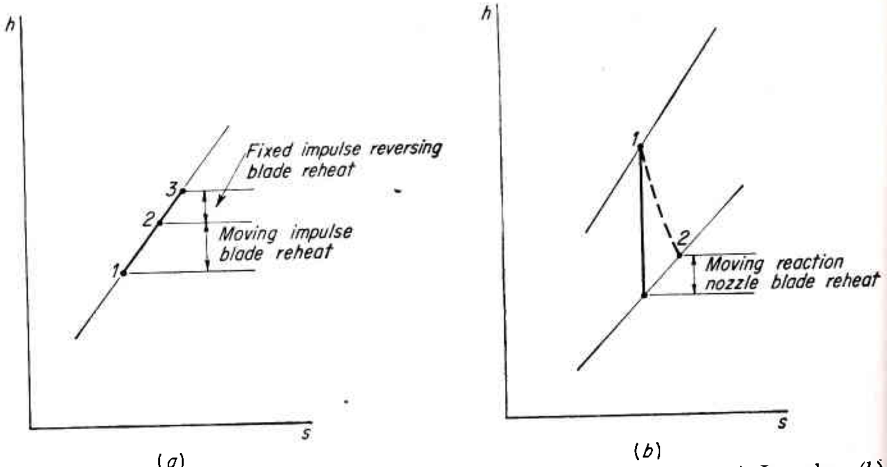


FIG. 12-8. *h-s* diagram for a turbine stage showing types of losses. (*a*) Impulse; (*b*) reaction.

The internal efficiency measures the departure of an actual turbine from ideal conditions,

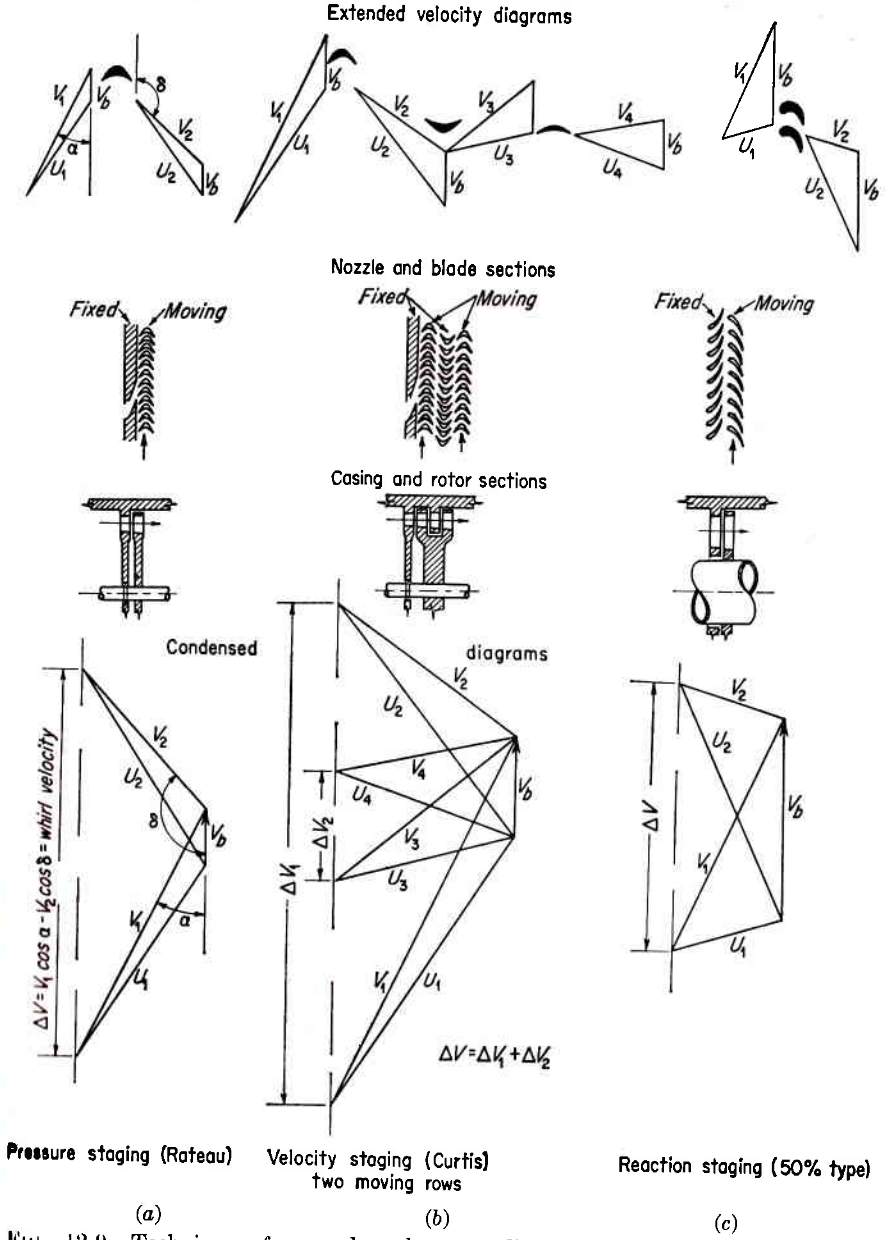
$$\eta_i = \frac{\Sigma W}{(\Delta H)_s} \tag{12-14}$$

where η_i = internal efficiency, fraction
ΣW = internal work delivered to the shaft inside the turbine
$(\Delta H)_s$ = isentropic enthalpy drop or over-all available energy

12-6. Vector Calculations. In turbine problems velocity vector diagrams are reduced to condensed forms (Fig. 12-9). The base-line vector represents blade velocity, and the steam-velocity vectors are superimposed on it (see Fig. 12-6). In pressure and velocity staging of actual turbines $U_2 < U_1$ in the moving blades and $V_3 < V_2$ in the reversing blades. For reaction blading $U_2 > U_1$ because of expansion, for 50 per cent reaction $U_2 = V_1$. The power output of the moving blades is

$$P = \frac{w}{g} \times \Delta V \tag{12-15}$$

where ΔV is the total tangential velocity change (whirl velocity) as in Fig. 12-9.



Pressure staging (Rateau)
(*a*)

Velocity staging (Curtis)
two moving rows
(*b*)

Reaction staging (50% type)
(*c*)

FIG. 12-9. Techniques for condensed vector-diagram constructions. (*a*) Single pressure stage; (*b*) two-row velocity stage; (*c*) reaction stage.

In the Curtis stage with two moving rows the first moving stage does considerably more work than the second. The small marginal return of three moving rows has made this arrangement fall from favor.

The Curtis stage with its large Δh drop makes it possible to drop the steam pressure and temperature quickly in the front end of the turbine. This reduces rotor length and leakage between stages and simplifies expansion problems. Usually Curtis stages are less efficient than the Rateau types and reaction stages more efficient at low pressures. The internal performance of a turbine is governed by: (1) nozzle and blade friction, (2) vector geometry—jet entrance and leaving angles; and (3) blade velocity ratio—(blade velocity)/(steam velocity). Entrance angles range closely about 15°. Decreasing blade exit angle (increasing γ of Fig. 12-6*a*) improves efficiency, but this has a closing-up effect on the blade passageway, which must have a given area to pass the steam flow. Ideally reaction and impulse turbines have equal efficiencies when run at their best blade velocity ratios; these range as in the following table.

Type	*Blade velocity ratio*
Pressure stage	0.4–0.45
Velocity stage (2-row)	0.2–0.28
Reaction stage	0.8–0.9

Actual blade efficiencies range between 78 and 88 per cent and depend on the blade factors discussed.

12-7. Turbine Types and Arrangements. Turbines are broadly classed as *condensing* or *noncondensing*, indicating that the back pressure is below or above atmospheric pressure. In *bleeder*, or *extraction*, turbines (Fig. 12-10) part of the steam leaves the casing before the exhaust. Removing small flows of steam for feedwater heating is usually called "bleeding." *Nonautomatic* and *automatic extraction* turbines used in industrial plants divert major parts of the throttle steam at intermediate stages of the turbine for process uses.

When waste or by-product steam at reduced pressures is available, it may be led into the lower stage of a turbine to do work; this is a *mixed-pressure* turbine. *Extraction-induction units* both remove and introduce steam into the turbine after the throttle.

The *reheat turbine* working in a reheat cycle returns its steam after partial expansion to the steam generator for resuperheating and then expands it to exhaust. *Superposed*, or *topping*, turbines are high-pressure noncondensing units installed in existing plants to exhaust into existing lower-pressure turbines. They increase plant capacity and improve over-all thermal performance.

Steam-turbine capacities range from less than 1 kw to over 800,000 kw. Inlet pressures range from a few pounds above atmospheric to 5,000 psig

and temperatures from saturated at the pressure to 1200 F. Speeds for generator drives are 3,600 and 1,800 rpm; geared units run up to 10,000 rpm and higher for small machines.

Figure 12-11 shows the turbine arrangements used to achieve the higher capacities. Maximum area of the exhaust annulus limits the steam flow that can be passed. Large annulus area requires long last-stage blades and buckets; the limit on length depends on the shaft speed, which sets up centrifugal forces in the moving buckets. Beyond certain forces the buckets would fail. Modern 3,600-rpm units may have 26-in. last-stage buckets and 1,800-rpm units 46 in. Single-casing units have a maximum capacity of about 33 mw. For larger capacities more than one exhaust annulus in parallel must be used, to pass the needed steam flow.

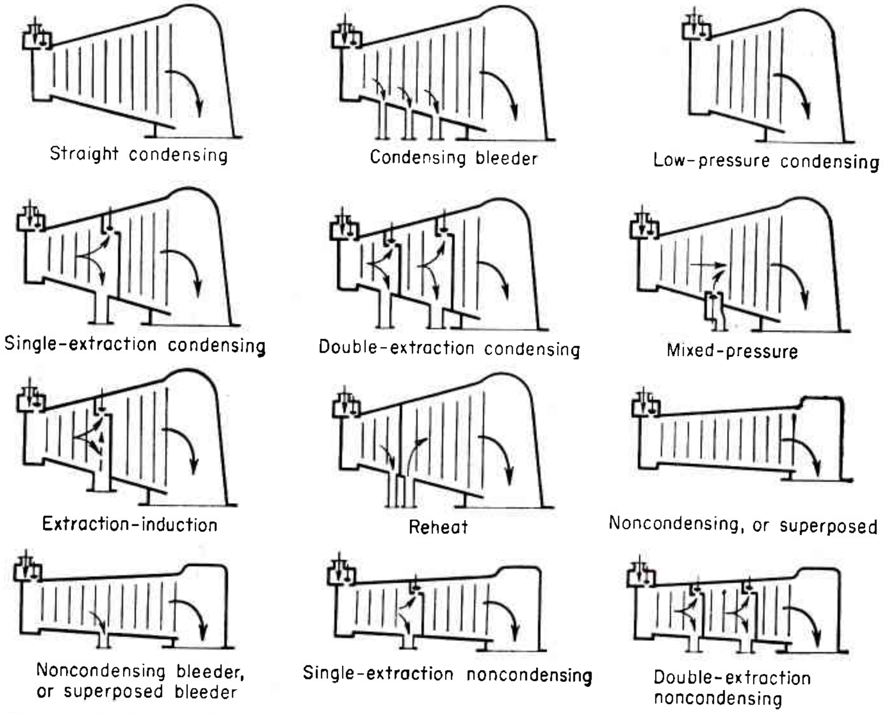
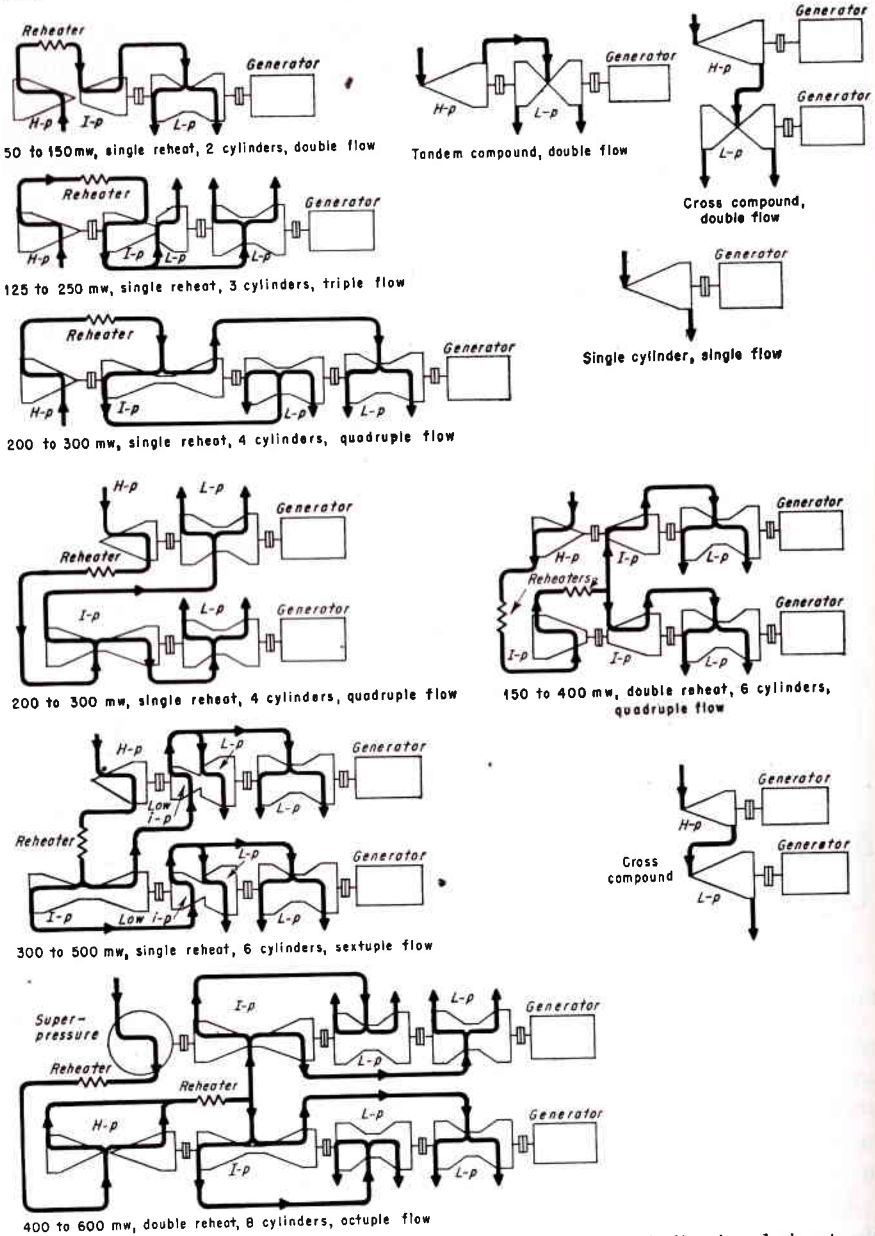

Straight condensing
Condensing bleeder
Low-pressure condensing
Single-extraction condensing
Double-extraction condensing
Mixed-pressure
Extraction-induction
Reheat
Noncondensing, or superposed
Noncondensing bleeder, or superposed bleeder
Single-extraction noncondensing
Double-extraction noncondensing

FIG. 12-10. Basic turbine types to meet a variety of power and process steam demands.

12-8. Mechanical-drive Turbines. Industry needs a wide variety of small- and medium-capacity turbines to drive pumps, compressors, blowers, fans, and other machinery. The turbine offers: (1) easily controlled variable speed, (2) quick starting, (3) freedom from spark hazard, and (4) adaptability to damp, hot, and other adverse conditions. More important, the turbine can remove energy from high-pressure steam and



50 to 150mw, single reheat, 2 cylinders, double flow

Tandem compound, double flow

Cross compound, double flow

125 to 250 mw, single reheat, 3 cylinders, triple flow

Single cylinder, single flow

200 to 300 mw, single reheat, 4 cylinders, quadruple flow

200 to 300 mw, single reheat, 4 cylinders, quadruple flow

150 to 400 mw, double reheat, 6 cylinders, quadruple flow

300 to 500 mw, single reheat, 6 cylinders, sextuple flow

Cross compound

400 to 600 mw, double reheat, 8 cylinders, octuple flow

FIG. 12-11. Typical plan-view arrangements of turbine units indicating design trends as unit capacity increases. Higher-capacity units indicate projected possible designs. (*Courtesy of Power.*)

make oil-free steam available at low pressures for other uses. Figure 12-12 shows a single-stage unit often used for mechanical drives. Capacities up to about 1,500 hp may be produced with 36-in.-diameter wheels. Speeds range from 900 to 10,000 rpm, with most running at 1,500 to 4,000 rpm.

Small multistage units designed for mechanical drive are often considered an intermediate class between mechanical drive and large-capacity generator drives. Pressure staging usually gives higher efficiency than single-stage units. These units are built up to about 10 mw, with speeds ranging from 1,000 to 5,500 rpm; they may be condensing or noncondensing and have bleed or extraction points. They may have a Curtis stage ahead of one or more Rateau stages (Fig. 12-13).

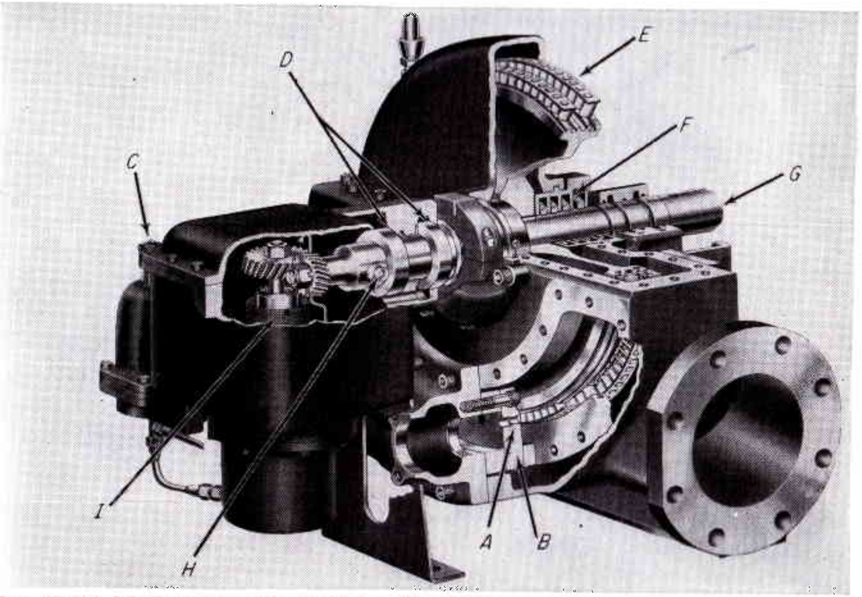


FIG. 12-12. Mechanical-drive turbine. Available from 10 to 1,200 hp; 30 per cent speed range with one two-velocity stage. *A*, nozzle; *B*, nozzle plate; *C*, hand speed changer; *D*, thrust runners; *E*, buckets and shroud bands; *F*, carbon packing rings and spring assembly; *G*, rotor; *H*, overspeed governor; *I*, oil-relay governing system. (*Courtesy of General Electric Company.*)

Reduction gears allow using high-efficiency high-speed turbines to drive low-speed loads. Gears also allow the turbine to drive two shafts at two different speeds.

12-9. Industrial-type Turbines. These units have capacities ranging from 500 to 10,000 kw and use steam of 600 to 1,450 psig pressure and 750 to 1000 F temperature. These multistage units run at 3,600 rpm in single casings. The industrial turbine fits a heat cycle furnishing both power and process steam.

Steam used for this double duty proves to be an economical energy arrangement. Turbines for this purpose are arranged as (1) noncondensing, with exhaust to process; (2) condensing or noncondensing, with controlled extraction of steam; and (3) condensing-induction, using by-product steam.

Figure 12-14 shows a noncondensing turbine with single automatic extraction. Figure 12-15 shows how these turbines would be used in an industrial cycle.

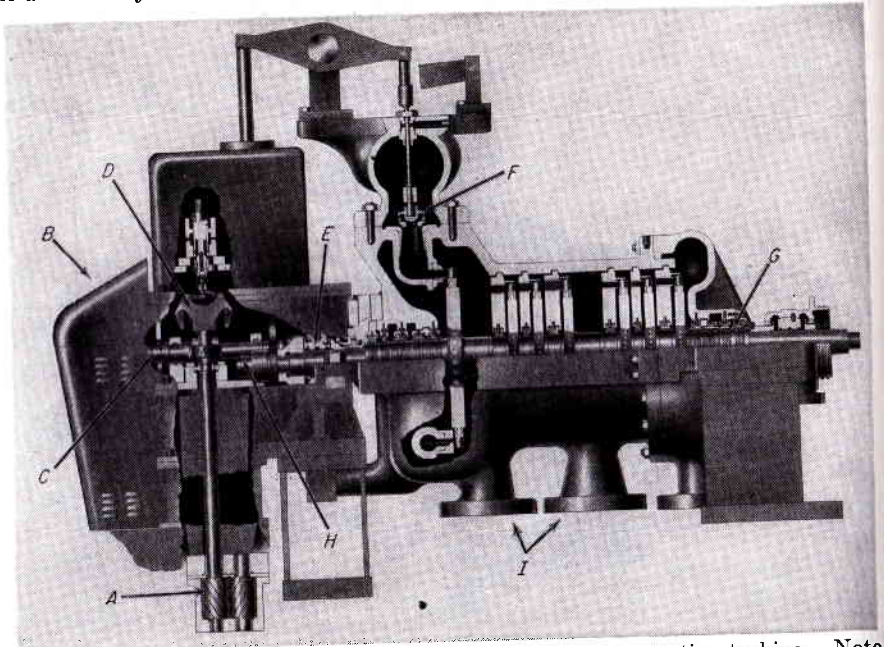


Fig. 12-13. Typical multistage mechanical-drive or power-generating turbine. Note the seven Rateau stages. *A*, oil pumps; *B*, instrument and control panel; *C*, turning gear and clutch; *D*, speed governor; *E*, tapered land thrust bearing; *F*, control valves; *G*, labyrinth packing; *H*, overspeed governor; *I*, extraction openings. (*Courtesy of General Electric Company.*)

12-10. Central-station Turbine Generators. In central stations the main turbine drives an electric generator. Capacities range from 16,500 to 800,000 kw, under a variety of steam conditions as in the following table. Pressures and temperatures up to 2400 psig and 1050 F are

Pressure, psig	*Temperature, °F*
850	900
1,250	950
1,450	1050/1000
2,400	1050/1000
3,500	1050/1050/1050
5,000	1200/1050/1050

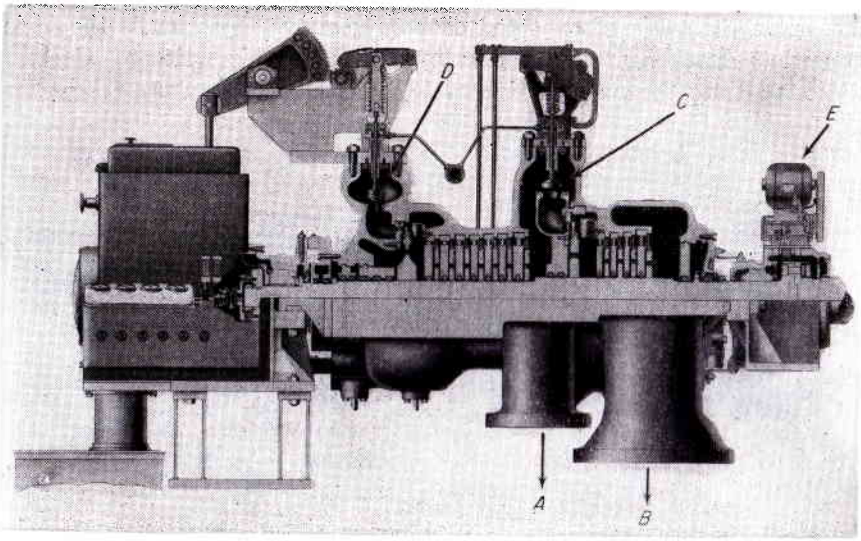


Fig. 12-14. Typical industrial turbine of the single automatic-extraction noncondensing type. *A*, controlled pressure extraction; *B*, principal turbine exhaust; *C*, extraction pressure-control valve; *D*, inlet speed-governing valve; *E*, turning-gear motor. (*Courtesy of General Electric Company.*)

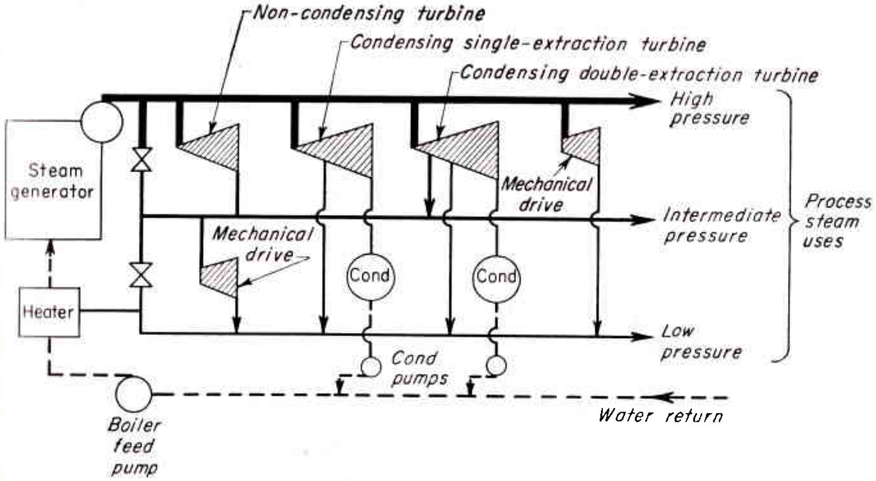


Fig. 12-15. Use of industrial turbines to supply process steam and generate by-product power.

chosen on the basis of expected economy; above these levels they are somewhat experimental and developmental at this writing. Central stations all use the regenerative cycle, and the majority use reheat for

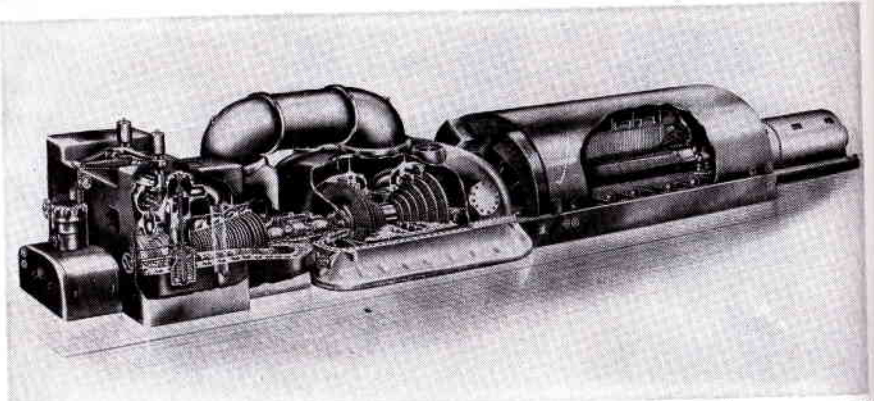


FIG. 12-16. Phantom view of large nonreheat steam turbine of the tandem compound type with high pressure and double-flow exhaust sections. Unit rated at 100 mw, 1,250 psig 950 F inlet conditions, and 1.5 in. Hg exhaust. *(Courtesy of Westinghouse Electric Corporation.)*

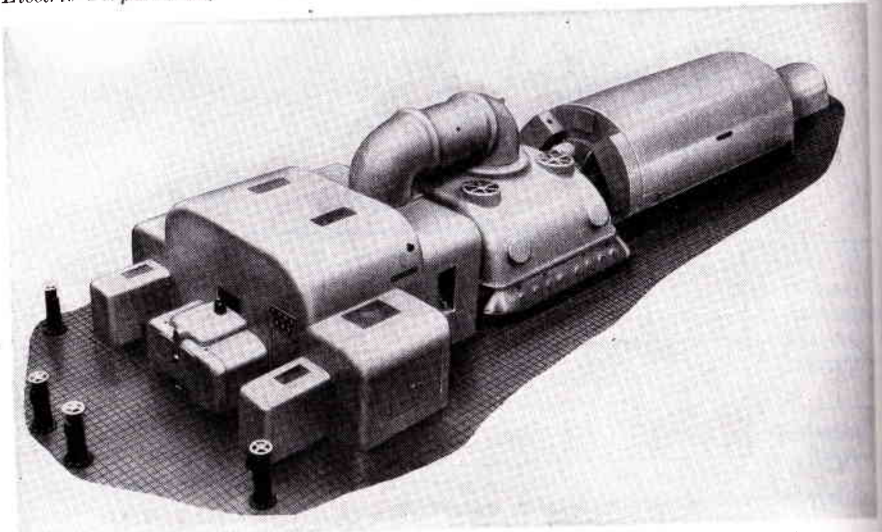


FIG. 12-17. Typical outside appearance of turbine-generator unit as installed in power station. *(Courtesy of Westinghouse Electric Corporation.)*

its efficiency advantage. The bleed steam pressures vary with the turbine load.

Figures 12-16 and 12-17 show some central-station turbine-generator designs of the nonreheat type. Figure 12-18 shows a reheat turbine generator with its main condenser.

12-11. Nuclear-power-station Turbines. Some of the nuclear plants built in the 1950's and 1960's use saturated steam at pressures up to 600 psig. The low pressures make the steam excessively wet in the low-pressure stages. These units have special moisture-catching devices in the stages to remove the moisture; in some units the entire steam flow passes through a separator before continuing to exhaust.

12-12. Steam-turbine Progress. Increasing throttle temperature and pressure has improved station thermal efficiency. The metal strength of

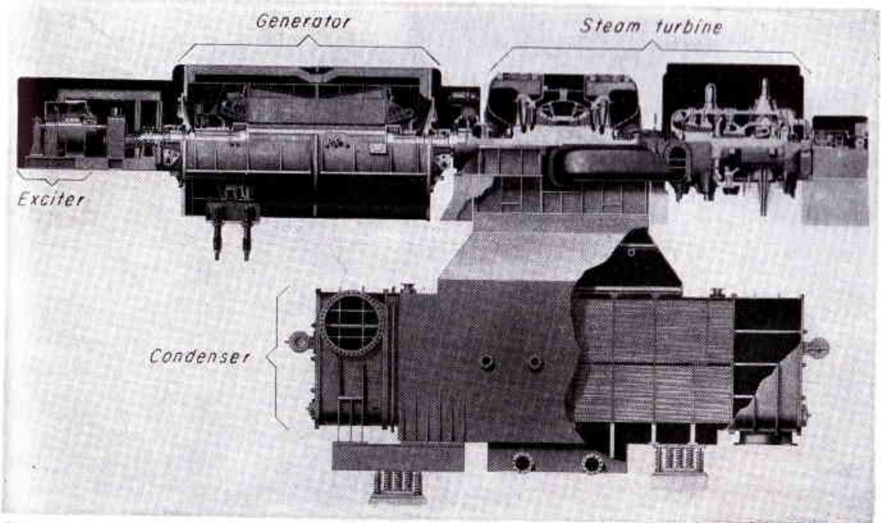


FIG. 12-18. Complete assembly of turbine, generator, and condenser for central-station use. Steam turbine is of the 100-mw-capacity reheat type, tandem compound arrangement with high pressure, intermediate pressure, and double-flow low-pressure cylinder. Note relative sizes of components. Condenser has 50,000 sq ft of heating surface. Turbine operates at 3,600 rpm. *(Courtesy of Allis-Chalmers Manufacturing Company.)*

those parts coming in contact with the steam limits the maximum temperature that may be used. While furnace flame temperatures range around 3000 F, the maximum steam temperature used commercially is 1100 F and in one unit 1200 F. Most units use temperatures of 1050 F and below. In addition to throttle steam conditions the station heat rate may be improved by increasing the number of bleed points, using one or more stages of reheat, increasing the capacity of the unit, and designing better steam-flow passages in buckets and nozzles. Figure 12-19 shows the trend in turbine performance since 1915, with predictions for the future.

12-13. Standardized Units. Turbine standardization promotes repetitive manufacture and lowers production costs. In 1939 the National Defense Power Committee adopted certain standards that were modified in 1943 by an ASME committee on 3,600-rpm turbines. Later a joint committee of the ASME and AIEE formed a complete standard for a selected range of turbine-generator ratings, published in 1945.

These standards helped: (1) lower first costs, (2) shorten delivery time, (3) standardize drawing layouts, (4) standardize performance data, (5) lower installation costs, and (6) standardize bleed-steam conditions. These standards do not restrict technical advances, since many of the larger units go beyond the standards range.

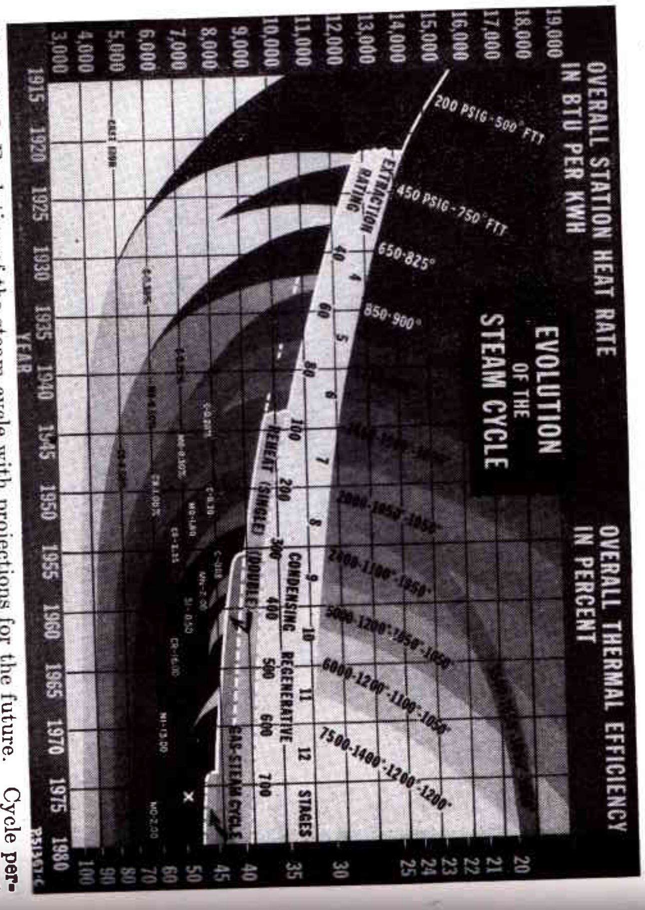


Fig. 12-19. Evolution of the steam cycle with projections for the future. Cycle performance, turbine ratings, extraction stages, and types of alloys used are indicated over a span of 65 years of turbine development. (*Courtesy of Westinghouse Electric Corporation.*)

Table 12-1 shows the latest *preferred standards* based on ASME Standard 100 and AIEE Standards 601 and 602. The generator capability varies with the cooling-hydrogen pressure.

While no standard exists for the larger machines, manufacturers attempt to market these to several companies simultaneously and so obtain the price advantage of mass production.

12-14. Turbine Governing. Governors control turbine-shaft speeds and also control the pressure of extracted steam in automatic-extraction units. Turbine speed directly affects the frequency of output of an a-c generator and must be kept within certain limits for satisfactory operation.

TABLE 12-1. PREFERRED STANDARDS FOR LARGE 3,600-RPM THREE-PHASE 60-CYCLE CONDENSING STEAM-TURBINE GENERATORS

	Air-cooled generator	Hydrogen-cooled generators rated for 0.5 psig hydrogen pressure									
Turbine rating,[a] kw	12,650	16,500	22,000	33,000	44,000		66,000		100,000[b]		150,000
Generator rating:											
Kva	13,529	17,647	23,529	35,294	47,058	47,058	70,588	70,588	106,951	106,951	160,428
Power factor	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85
Short-circuit ratio	0.80	0.80	0.80	0.80	0.80	0.80	0.80	0.80	0.8	0.8	0.8
Throttle pressure, psig	600	850	850	850	850	1,250	850	1,250	1,450	1,450[c]	1,800[c]
Throttle temperature, °F	825	900	900	900	900	950	900	950	1,000	1,000	1,000
Reheat temperature, °F	......	......	......	......	......	......	......	......	......	1,000	1,000
Number of extraction openings	4	4	4	5	5	5	5	5	5	5	7
Saturation temperatures at openings at "turbine rating" with all extraction openings in service, °F:											
1st	180	180	180	180	......	180	......	180	185	180	150
2d	240	240	240	240	......	240	......	240	250	245	185
3d	290	290	290	290	......	290	......	290	310	305	235
4th	360	360	360	360	......	360	......	360	385	375	280
5th	......	......	......	420	......	420	......	420	450	450	325
6th	......	......	......	......	......	......	......	......	......	......	375
7th	......	......	......	......	......	......	......	......	......	......	455
Exhaust pressure, in. Hg abs	1.5	1.5	1.5	1.5	......	1.5	......	1.5	1.5	1.5	1.5
Generator capability at 0.85 power factor and 15 psig hydrogen pressure, kva	......	20,294	27,058	40,588	54,117		81,176		122,994		184,492
Generator capability at 0.85 power factor and 30 psig hydrogen pressure, kva	......	22,059	29,411	44,118	58,822		88,235		133,689		200,535

A tolerance of 10 F shall apply to above saturation temperatures. Tolerances shall be unilateral, if possible, so as not to reduce the spread in temperature between adjacent extraction openings.

[a] The "turbine rating" is guaranteed continuous output at generator terminals when the turbine is clean and operating under specified throttle steam pressure and temperature and 2.5 in. Hg abs exhaust pressure, with full extraction from all extraction openings.

[b] These are two different turbines; the first for regenerative-cycle operation, and the second a machine for reheat-regenerative-cycle operation.

[c] A 10 per cent pressure drop is assumed between high-pressure turbine exhaust and low-pressure turbine inlet for the reheat machines.

A governor controls shaft speed and extraction pressures by varying the opening of the inlet and bypass valves with variation in mechanical or electric load. Figure 12-20 shows the schematic arrangements used. The primary element, either a *mechanical-centrifugal* or a *hydraulic* device, responds to speed or pressure variations.

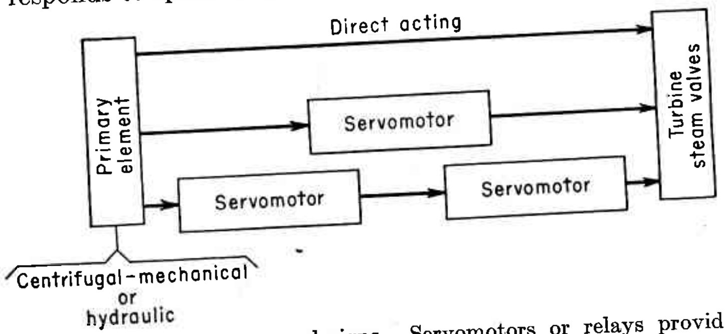


FIG. 12-20. Elements of governor designs. Servomotors or relays provide large operating forces, while primary element retains its sensitivity.

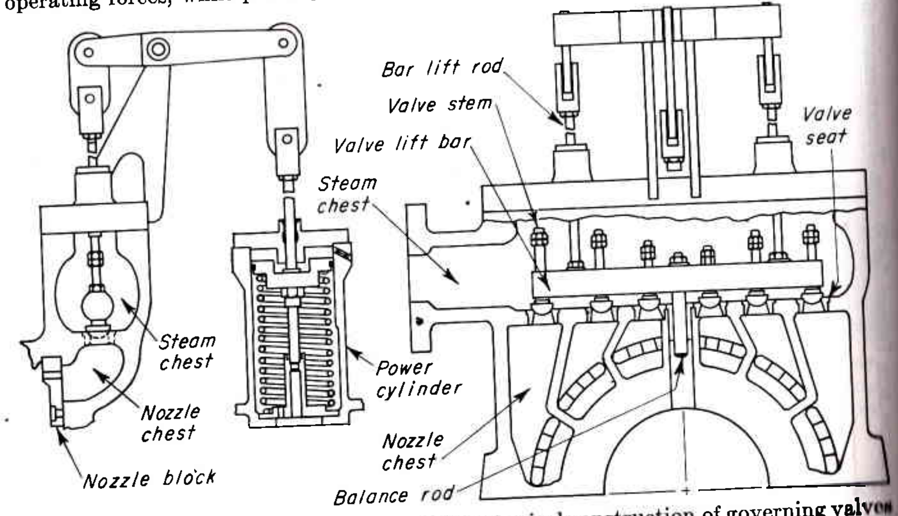


FIG. 12-21. Cross-sectional drawing showing a typical construction of governing valves and operating mechanism. (*Courtesy of Allis-Chalmers Manufacturing Company.*)

In small mechanical-drive units the primary element may position the steam valves directly. A primary element usually does not generate enough force to move valves, and so some form of power-step-up device is used. In most units the primary element signals a servomotor, which in turn controls the force developed by an outside energy source (usually a pressurized fluid) to move the steam valves.

While small turbines use straight throttling governing, this method introduces large losses in handling large steam flows. Figure 12-21 shows a more efficient nozzle-control arrangement; here the governor opens and closes several steam-admission valves in sequence. The system uses lift rods or cams to open the individual nozzle valves. This arrangement controls steam flow by opening additional nozzle area rather than primarily throttling steam—throttling, however, can never be completely avoided. In some larger units steam enters the second and third stages to develop maximum rated capacity.

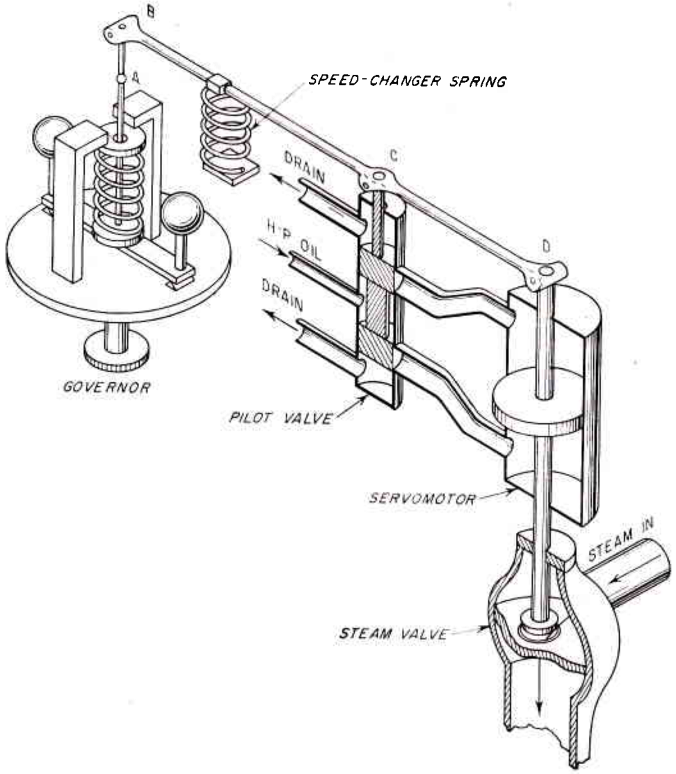


FIG. 12-22. Schematic sketch of governor system with flyball type of centrifugal primary element controlling pilot valve, which in turn operates servomotor and steam valve.

Figures 12-22 and 12-23 show the two basic governing systems in schematic form. Each design has a speed changer that may be operated manually or automatically. These governors have a drooping speed characteristic; the droop may be some 4 to 10 per cent of the no-load speed. The mechanical-drive turbines use the higher droops.

Figure 12-24 shows typical *speed-regulation* curves for two units *A* and *B*. These give general trends; actual characteristics will be curved and will have discontinuities. The dashed lines about *A* show that the speed changer shifts a regulation curve parallel to itself. Hence at constant load the speed changer may alter speed (and frequency) from S_1 to S_2. Different loads may be carried at constant speed by changing the setting from L_1 to L_2.

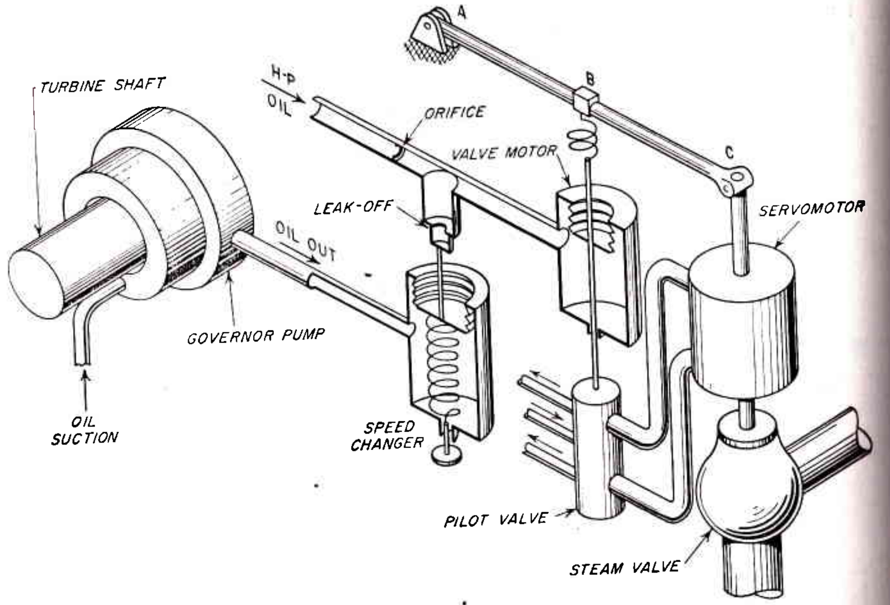


FIG. 12-23. Schematic sketch of hydraulic governor system. Governor pump pressure regulates speed-changer bellows, which control valve motor, setting pilot valve as needed. Pilot valve regulates steam flow through servomotor.

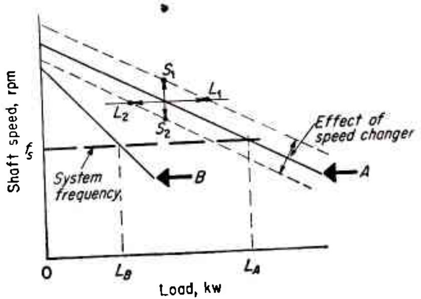


FIG. 12-24. Turbine speed-regulation curves. Speed changer shifts curve about parallel to itself. This makes changes in *load*, *speed*, and *load division* feasible.

Two or more a-c turbine generators connected in parallel run at a common synchronous speed. In Fig. 12-24 at system frequency f_s the units carry a total load equal to $L_A + L_B$. Using the speed changers on A and B the units may be operated at: (1) increased total load and constant frequency, (2) decreased total load and constant frequency, and (3) different load divisions between units at any total load and frequency.

Bleed steam from a regenerative-cycle turbine is self-regulating. Steam can flow only to the heaters, as the feedwater condenses it in the connected heater.

Automatic-extraction turbines have complicated governing systems consisting of a speed-responsive mechanism and a pressure-responsive element to keep extraction pressure constant as both extraction flow and turbine shaft load vary. Double automatic-extraction turbines must have two pressure elements, one for each extraction point.

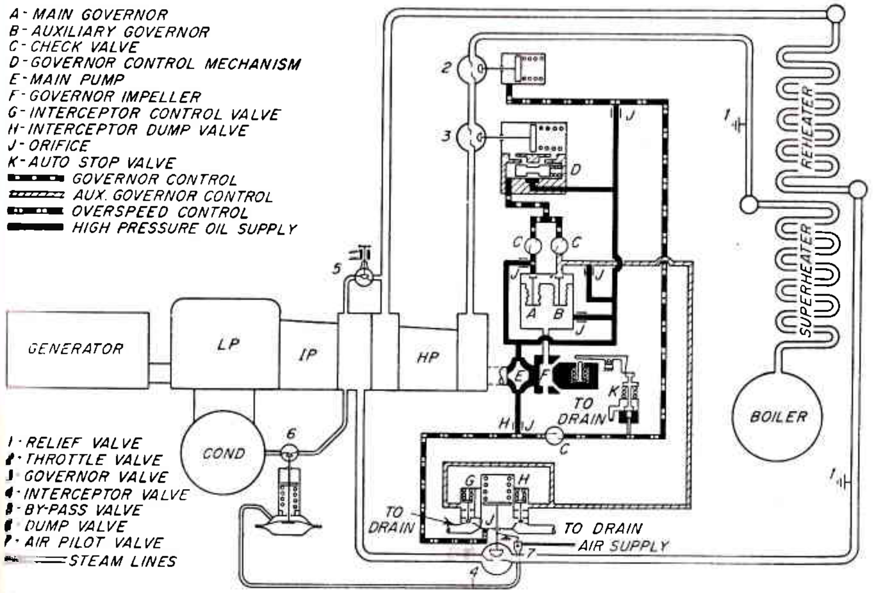


FIG. 12-25. Typical control diagram for a reheat turbine. (*Courtesy of Westinghouse Electric Corporation.*)

Turbine control has other devices such as the overspeed trip, low-oil-pressure trip, steam-flow-limiting device, and load-drop anticipator. These arrangements attempt to foresee every emergency condition that might arise and provide automatic protection against them.

The piping and reheater used in a reheat cycle complicate the control problem of the turbine. Figure 12-25 shows a hydraulic type of governor system for a reheat turbine; it has an auxiliary governor, interceptor valve, bypass valve, and dump valve.

Bypass valve 5 avoids overheating the intermediate- and low-pressure units during starting procedures by mixing some of the exhaust from the high-pressure turbine with reheated steam. When the interceptor valve 4 closes on load loss, the dump valve 6 prevents steam leaking through the high-pressure-turbine exhaust and intermediate-pressure-turbine inlet seals and overspeeding the turbine.

Main governor *A* controls speed and loading for normal operations. When speed increases suddenly from a large load loss, the auxiliary governor takes control. It closes interceptor valve *G* and control valve 3 at about 3 per cent excess speed. This shuts off steam flow to both high-pressure and intermediate-pressure turbines from the boiler and reheater. The dump valve opens simultaneously, passing leakage steam to the condenser to prevent the overspeed trip from closing throttle valve 2. At speeds in the control range the auxiliary governor then closes the dump valve and sets the interceptor valve to throttle the steam trapped between the control and interceptor valves until the steam pressure drops to a safe level. It finally opens the control valves enough and the interceptor valve wide to maintain the 3 per cent higher speed. The unit may then be reloaded as soon as system conditions permit.

12-15. Turbine Generators. Steam turbines drive a-c synchronous generators (alternators) of two- or four-pole design. These are three-phase machines offering economic advantages in generation and transmission. In the United States the standard frequency is 60 cycles, which means that four-pole generators must turn at 1,800 rpm and two-pole at 3,600 rpm.

For generators rated 500 kw and higher and using 600 volts and higher the rotor carries the field and the stator the armature winding. This makes the slip rings for the rotor winding carry only moderate currents at relatively low voltages. Armature external leads carrying high voltages and currents can be better braced and insulated when stationary. The large, heavily insulated armature conductors can be better supported to withstand heavy short-circuit currents.

Generator losses appearing as heat must be constantly removed to avoid damaging the windings. Room air cools small salient-pole generators; fan blades on the rotor induce air flow over the windings.

Large generators have cylindrical rotors with a minimum of heat-dissipation surface, and so they must have forced ventilation to remove the heat. Practically all large generators use an enclosed system with *air* or *hydrogen* coolant. The gas picks up heat from the generator windings and gives it up to circulating water in a heat exchanger; the water may be main condensate or raw water. Table 12-1 shows the effect of increased hydrogen pressure, which raises the heat-removal capacity. Raising hydrogen pressure from 15 to 30 psig increases generator capability as much as 10 per cent.

Hydrogen reduces windage loss to about 10 per cent that of air. The higher thermal conductivity lets generators develop about 25 per cent more output than with air. Insulation stands up better when not exposed to the oxygen in air. Hydrogen-cooled generators must be gastight and explosion-proof. Labyrinth-type oil seals supplied with oil at 5 psi pressure above the hydrogen keep leakage negligible. Automatic controls maintain hydrogen pressure, temperature, and purity. Impurities in the H_2 increase circulating-fan power and the hazard of explosion; it is kept at about 95 to 99 per cent purity. Carbon dioxide (CO_2) purges the system of air before filling it with hydrogen and also purges it of H_2 before opening up for maintenance. The gas-control system feeds 99.6 per cent pure H_2 to the generator and continuously scavenges gas contaminants to preserve purity.

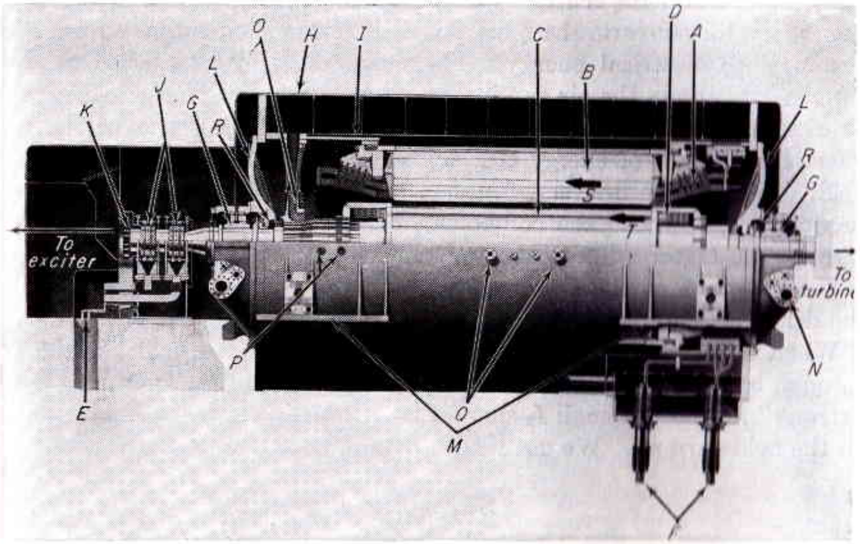


FIG. 12-26. Section through a typical fully supercharged generator. This particular design is used in units of 150 mw and up. Basic elements are: *A*, stator windings; *B*, stator core; *C*, generator rotor (field); *D*, field windings; *E*, field leads; *F*, main terminals (hydrogen jet cooled); *G*, spherical seated bearings; *H*, outer-yoke double-shell construction (note that hydrogen coolers not shown are located in yoke); *I*, inner yoke (note that outer yoke is rigid and inner yoke must be able to flex owing to ellipsoidal deformation of coils and structure during generation); *J*, collector rings; *K*, collector-ring cooling fan; *L*, end bells; *M*, mounting feet; *N*, bearing oil connection; *O*, circulating blower for hydrogen coolant; *P*, hydrogen-cooler water connections; *Q*, hydrogen connections for charge and removal of coolant; *R*, thrust type of hydrogen seal; *S*, hydrogen coolant flows through ducts embedded in stator turns and through cooling holes in laminations; *T*, hydrogen cools rotor by direct copper-to-hydrogen heat exchange. (*Courtesy of Allis-Chalmers Manufacturing Company.*)

Until 1952 generators circulated the coolant about the insulation exterior in vent ducts. Modern units have blowers on the rotor to force the H_2 through hollow ducts in the stator and rotor copper at high velocity. This direct metal-to-gas heat transfer reduces the temperature differential and allows raising the generator rating for a given size. Direct liquid cooling inside the conductor ducts on stators has been the latest step in uprating generators, because of the better heat-transfer characteristics of a liquid film compared with gas.

Generators must have a separate d-c generator to supply their field windings. Some units have an exciter connected on the same shaft; others use separately motor-driven exciters. Figure 12-26 shows a modern generator for central-station service.

A generator converts the mechanical shaft energy it receives from the turbine into electrical energy. The magnetic field established in the rotor passes across the air gap between rotor and stator and must sweep over the armature conductors. Any factors varying the excitation affect the amount of energy transmitted through this moving magnetic field. The air-gap flux is a function mainly of the field current (flux-producing effect) and the component of armature current that lags at zero power factor (flux-weakening effect). At all times the gap flux is the resultant of these two effects. In the short-circuited generator the demagnetizing component or armature current dominates.

When a generator is short-circuited, we can send a field current I_b through its rotor winding to produce *rated* current in the armature. In a "strong" machine a small I_b shows that the air-gap flux depends mainly on the field current. We use this to determine the *short-circuit* ratio

$$SCR \doteq \frac{I_a}{I_b} \qquad (12\text{-}16)$$

where I_a = field current to produce rated voltage at no load and rated frequency
I_b = field current to produce rated current in short-circuited armature

The stability of a power system depends on the air-gap flux of the generators during a short circuit or sudden change in load. High short-circuit ratios indicate inherent generator stability but increase the generator investment. In contrast low short-circuit-ratio generators with automatic voltage control cost less. The short-circuit ratio for most generators varies from about 0.6 to 0.9.

12-16. Turbine Performance. Turbine behavior can be expressed by: (1) the steam-flow process through the unit—expansion line or condition curve; (2) the steam-flow rate through the unit; (3) comparison with the ideal—engine efficiency; (4) absolute performance—heat rate or thermal efficiency; and (5) losses—exhaust, mechanical, generator, radiation, etc.

Expansion lines on the Mollier chart show the steam conditions throughout the turbine from its initial throttle condition to the exhaust pressure. Being an irreversible expansion, the process cannot be shown accurately. For practical performance calculations the line is assumed to be the locus of extraction steam points.

Figure 12-27*a* shows the basic features of an expansion line that is considered to be adiabatic. The manufacturer normally supplies curves describing the machine performance. Expansion lines can be computed from the concepts of Figs. 12-7 and 12-8 for all turbines. Based on Fig. 12-27*a*, the internal efficiency for a nonextraction turbine is

$$\eta_i = \frac{h_1 - h_2}{h_1 - h_{2'}} \qquad (12\text{-}17)$$

where η_i = internal efficiency
h = enthalpies as in Fig. 12-27*a*

To compare actual and ideal performance we use engine efficiency. For a nonextraction turbine

$$\eta_e = \frac{h_1 - h_2}{h_1 - h_{2'}} \times \eta_g \times \eta_m = \frac{3{,}413}{w_s(h_1 - h_{2'})} \qquad (12\text{-}18)$$

where η_e = over-all engine efficiency
η_g = electrical generator efficiency
η_m = turbine mechanical efficiency
w_s = steam rate, lb per kwhr

These formulas must be modified for the different types of turbines. The ASME Power Test Code of Steam Turbines gives these various relations, for example, for a complete-expansion or nonextraction turbine:

$$\eta_e = \frac{3{,}412.7(P_g \text{ or } P_c)}{(W_1 - \Sigma W_g)(h_1 - h_{sx}) + \Sigma W_g(h_1 - h_{sg2})} \qquad (12\text{-}19)$$

For a regenerative extraction turbine

$$\eta_e = \frac{3{,}412.7(P_g \text{ or } P_c)}{\Sigma[W_{b2}(h_1 - h_{sb2}) + W_{b3}(h_1 - h_{sb3}) + \cdots + W_4(h_1 - h_{s4})] + \Sigma W_g(h_1 - h_{sg2})} \qquad (12\text{-}20)$$

where η_e = over-all engine efficiency
P_c or P_g = turbine coupling or turbine generator output, kw
W_1 = flow of steam entering turbine, lb per hr
ΣW_g = gland steam not returning to system, lb per hr
W_{b2}, W_{b3} = steam extracted at bleeder points, lb per hr
W_4 = exhaust steam flow, lb per hr
h_1 = enthalpy of steam supplied, Btu per lb
h_{sx} = enthalpy of exhaust steam at initial steam entropy, Btu per lb
h_{sg2} = enthalpy of gland loss steam at leak-off pressure and entropy of supplied steam, Btu per lb
h_{sb2}, h_{sb3}, h_{s4} = enthalpy of steam at extraction and exhaust pressure and entropy of supplied steam, Btu per lb

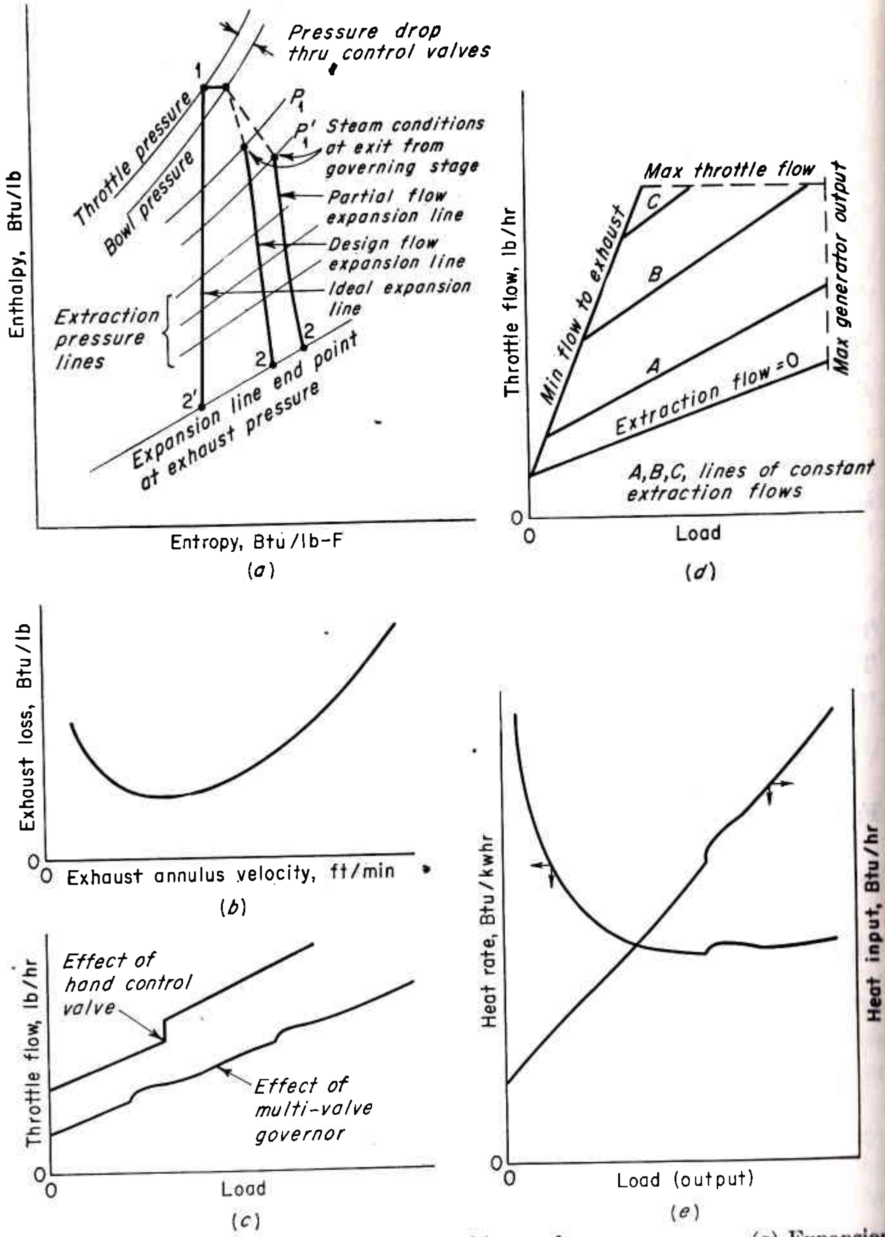


FIG. 12-27. General trends of principal turbine performance curves. (*a*) Expansion or condition line; (*b*) exhaust loss; (*c*) steam-flow (Willans line) curve for nonextraction machines; (*d*) steam-flow curve for single automatic-extraction machines; (*e*) heat rate and input-output curve. In all cases specific data must come from the manufacturer or other sources, as the foregoing is intended only to illustrate trends.

Engine efficiencies range from 30 to 50 per cent for small units and from 80 to 87 per cent for large utility units.

Turbine performance data include mechanical and generator losses. Equation (12-17) shows them as efficiencies, and so energy losses will have to be converted to this form. Mechanical losses include bearing losses, oil-pump losses, and frequently generator bearing losses. Generator losses include all mechanical and electrical losses of the generator expect bearing losses if these are included in mechanical losses.

Leaving losses of the steam exhausted from the last stage are analyzed too. They include the kinetic energy of the steam as it leaves the last stage, the pressure drop from the exit of the last-stage buckets to the condenser flange, and the losses from insufficient flow at low loads. Manufacturers give these data in curve form as in Fig. 12-27*b*. Since the expansion line is drawn to the condenser flange pressure, the exhaust loss from the curve (Fig. 21-27*b*) is added to the enthalpy from the expansion-line terminal point to get the actual total enthalpy of steam leaving the last-stage turbine buckets.

Steam-flow curves, or Willans lines (Fig. 12-27*c*), describe the steam-flow variation with load, for mechanical-drive and straight nonextraction turbines. The mechanical-drive turbine often has a second nozzle set opened manually for loads above half load; these cause a discontinuity in the curve. Multivalved turbines have discontinuities when each successive valve opens.

Figure 12-27*d* shows a typical Willans-line family for a double automatic-extraction turbine. Curves are plotted for given values of extraction flow. Some steam must always flow between the extraction point and the exhaust to keep buckets from overheating from the windage loss. The limit line for minimum flow to exhaust indicates this; note the other limits plotted.

The turbine heat rate gives the amount of heat energy needed by the unit to develop one unit of output, the kilowatthour, given as Btu per kilowatthour. Output may be measured at the turbine coupling or the generator terminals. The latter measurement gives the *combined turbine-generator heat rate*. Dividing this by the steam-generator efficiency gives the *station heat rate*. Figure 12-27*e* shows the input-output and heat-rate curves for a turbine with a two-valve governor control. The ASME Power Test Code outlines the method of computing heat rates; for example, for a complete expansion turbine

$$HR = \frac{(W_1 - W_g)(h_1 - h_{fx}) + W_g(h_1 - h_{fg2})}{P_g \text{ or } P_c} \tag{12-21}$$

For a regenerative-cycle turbine

$$HR = \frac{W_1(h_1 - h_8)}{P_g \text{ or } P_c} \tag{12-22}$$

where some symbols have been previously explained and where

HR = turbine or turbine-generator heat rate, Btu per kwhr
h_{fx} = saturated liquid enthalpy at exhaust pressure, Btu per lb
h_{fg2} = saturated liquid enthalpy at gland leak-off pressure, Btu per lb
h_8 = enthalpy of feedwater leaving last heater, Btu per lb

This equation applies to a strictly regenerative turbine; consult the Code for allowances such as extraneous leak-off steam from glands, condenser leakage, and condensate from steam-jet air pumps.

To convert heat rate to thermal efficiency,

$$\eta_t = \frac{3,413}{HR} \qquad (12\text{-}23)$$

Turbine heat rates vary widely with operating conditions of inlet, reheat, and exhaust steam, turbine size, and the cycle. Table 12-2 gives a brief survey of turbine generators and their heat rates.

TABLE 12-2. REPRESENTATIVE TURBINE-GENERATOR HEAT RATES

Load, kw	Throttle pressure, psig	Throttle temperature, °F	Reheat temperature, °F	No. of extraction points	Exhaust pressure, in. Hg	Heat rate, Btu per kwhr
12,650	600	825	....	4	1.0	10,205
22,000	850	900	....	4	1.0	9,585
33,000	850	900	....	5	1.0	9,440
33,000	850	900	....	5	2.5	9,660
44,000	850	900	....	5	1.0	9,275
44,000	1,250	950	....	5	1.0	8,900
66,000	850	900	....	5	1.0	9,270
66,000	1,250	950	....	5	1.0	8,915
66,000	1,250	950	....	4	1.0	9,000
100,000	2,400	1100	1100	5	1.5	7,720
100,000	1,800	1050	1050	5	1.5	7,960
100,000	1,250	950	950	5	1.5	8,400
150,000	2,400	1100	1100	5	1.5	7,640
150,000	1,800	1050	1050	5	1.5	7,880
150,000	1,250	950	950	5	1.5	8,340
200,000	2,400	1100	1100	5	1.5	7,590
200,000	1,800	1050	1050	5	1.5	7,850
200,000	1,250	950	950	5	1.5	8,330
250,000	2,000	1000	1000	7	1.5	7,685
300,000	2,000	1000	1000	7	1.5	7,670
350,000	2,400	1000	1000	7	1.5	7,630

12-17. Turbine Supervisory Instruments. Modern high-speed turbines must be precision-built to maintain the small clearances between rotor and stationary parts. During starting and stopping the turbine alternately heats and cools, and clearance must be maintained while the parts expand and contract. To ensure continuous operation, the modern turbine has supervisory instruments to check on: (1) rotor vibration, (2) rotor position, (3) differential (rotor-casing) expansion, (4) rotor eccentricity, (5) casing expansion (axial), and (6) valve position and speed. Figure 12-28 shows the instruments and their location on a unit.

12-18. Turbine Purchasing. Turbine specifications should give:

Rating. Turbine kilowatts, generator kilovolt-amperes, power factor, short-circuit ratio, generator voltage, phases, frequency, and exciter characteristics.

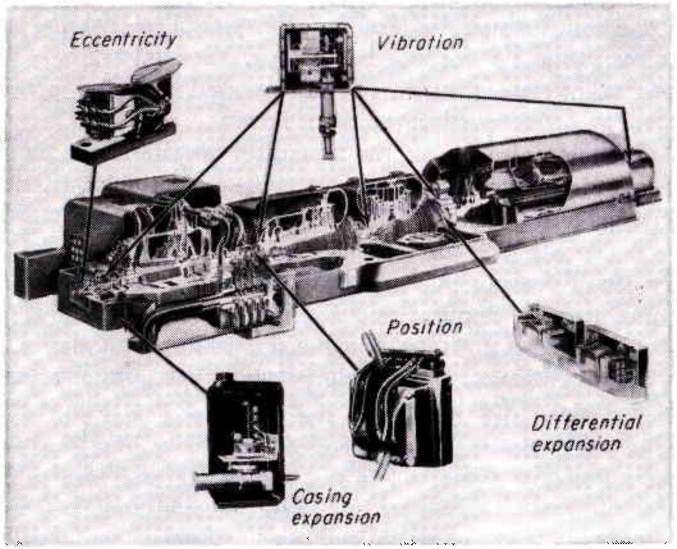


FIG. 12-28. General location and types of supervisory instruments for turbine operation. Their indications are commonly recorded on charts mounted on a recording panel. *(Courtesy of Westinghouse Manufacturing Corporation.)*

Steam conditions. Initial steam pressure, temperature, reheat pressure and temperature, exhaust pressure.

Steam-extraction arrangements. Number, pressure, steam flow, for automatic or nonautomatic extraction.

Air or hydrogen cooler. Temperature and pressure.

Bill of material. The turbine is usually furnished with a standard set of accessories (stop and throttle valve, tachometer, turning gear, etc.); other than standard furnishing must be indicated.

Shipping requirements. Weights and over-all dimensions as shipped.

Terms of payment. Usually progressive, about 60 per cent paid while construction proceeds and 40 per cent shortly after delivery.

Statement of escalation. Agreement for labor and material cost changes as construction progresses.

The standard price lists of a manufacturer give turbine-generator costs as a base price for given rating and steam conditions. Special accessories and features cost extra as shown by a separate price list. During periods of fast cost changes prices may be adjusted.

$$P_p = P_s \times C_1 \times C_2 \tag{12-24}$$

where P_p = present cost, dollars
P_s = cost from price list
C_1, C_2 = correction factors for changing labor or material costs

12-19. Turbine Testing. Manufacturers give their turbines functional tests, but not full loads. These may be governor, oil system, overspeed, and others. They usually are not shop-tested at rated pressure, but they are run at rated speed with steam flow.

Functional tests also check the generator in the shop, for example, the tightness of the hydrogen system. After erection the builder tests overspeed trips and checks all safety devices. The owner supplies labor to run a heat-rate test under the manufacturer's supervision. The builder is usually responsible for the unit during the acceptance test. The manufacturer accepts responsibility for alterations if the performance guarantee is not met, which seldom happens today.

12-20. Steam Engines. The reciprocating steam engine was the first type of prime mover used in power plants, but today it has been largely superseded by the turbine. In its most efficient forms the engine has a better heat rate than steam turbines, at capacities less than 1,000 hp.

Steam engines can be reversed and run efficiently at variable speed. For certain applications the steam engine still offers economic advantages.

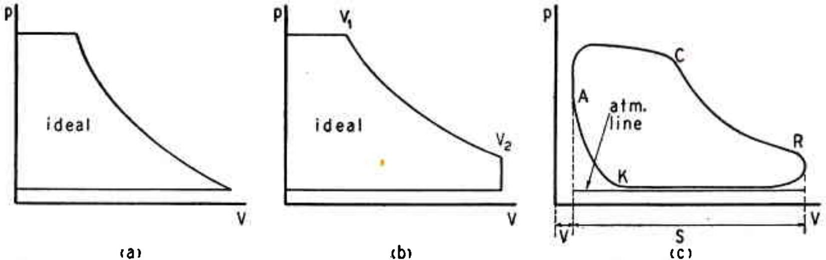
Steam engines may be classed as: (1) single-valve counterflow, (2) four-valve counterflow, or (3) uniflow. Any of these three types may be further classified: (1) as simple or compound; (2) as cutoff or throttle-governed; (3) as condensing or noncondensing; (4) by type of valve—D, piston, poppet, Corliss, etc.; (5) according to speed; or (6) as horizontal or vertical.

12-21. Indicator Diagram. An indicator makes it possible to study how steam expands in a cylinder as it acts upon or is acted upon by the piston. The indicator makes an automatic record of the variation in cylinder pressure with volume, a *P-V* diagram.

Figure 12-29 compares ideal and actual indicator cards. In (*a*) we have a *complete-expansion engine* and in (*b*) an *incomplete-expansion engine.* In (*c*) we have the card of an actual engine. The piston reciprocates through a stroke *S*. Port space needs and avoiding contact of piston and cylinder head dictate the requirement of a clearance volume *V′*.

The opening valve admits steam to the cylinder at *A* as the piston completes its exhaust stroke. Cylinder pressure quickly builds up to almost steam pressure at the inlet; as the piston reverses, steam flows in (admission) at about constant pressure to the cutoff point *C*, where the valve closes. Frictional and throttling pressure losses cause a small drop in steam pressure during admission.

The steam expands to a lower pressure from *C* to *R*, where the valve opens to allow steam to exhaust from the cylinder (to atmosphere in this example). The returning piston ejects the remaining steam at a pressure slightly above atmosphere because of throttling and friction in the exhaust valve.



(a) Complete expansion card (b) Incomplete expansion card (c) Practical card

FIG. 12-29. Reciprocating-engine indicator cards.

To cushion the piston and lessen inlet throttling losses, the release valve closes at *K* during the piston exhaust stroke. The trapped steam rises in pressure until the admission valve opens at *A*.

Although throttling causes some efficiency loss, the major loss is caused by cylinder condensation and reevaporation. The metal of the cylinder, piston, and valves alternately heats and cools as they contact hot inlet and cold exhaust steam. During admission, from *A* to *C*, some of the incoming steam condenses instead of working against the piston face. During the first part of process *C* to *R* some steam will be still condensing. As the steam expands it cools below the piston and cylinder temperature so that the low-pressure steam is partly reheated. But most of this heat energy is lost in the exhaust steam.

Steam engines may exhaust at a range of pressures from below to above atmospheric pressure. This will place the atmospheric line in different positions on the indicator card (Fig. 12-29*c*). The length of the atmospheric line is proportional to the piston stroke. The pressures can be calculated from the spring calibration. These data help locate the *P-V* axis on the card. These cards prove very useful in properly setting the engine valves.

Designers have established *diagram factors* based on actual experience that enables them to estimate closely the indicated horsepower (ihp) and mean effective pressure (mep) of actual engines for given speed, engine type, and inlet- and exhaust-steam pressures. These factors are

$$\text{Diagram factor} = \frac{\text{mep (actual indicator card)}}{\text{mep (ideal indicator card)}} \qquad (12\text{-}25)$$

Steam-engine performance can be improved several ways; by compounding the steam may expand in several cylinders in sequence. High-pressure steam enters the first cylinder, expands partly, and then exhausts to a second cylinder. The steam expands further in the second cylinder and may exhaust to waste or to a third cylinder. As many as four cylinders have been used by engines. These engines could develop large output but lost favor when steam turbines were developed with better efficiencies.

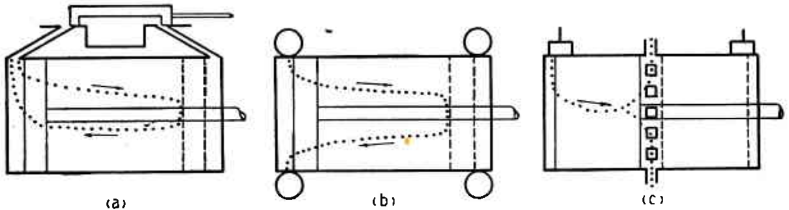

(a) Single valve counterflow (b) Four valve counterflow (c) Uniflow

FIG. 12-30. Reciprocating-engine steam action within engine cylinder.

12-22. Steam-engine Governing. Figure 12-29*c* gives a clue to the duties required of a governor. The card area must be varied to vary the load and maintain a constant shaft speed.

Cutoff-governed engines do this by advancing or delaying cutoff *C* during the expansion or working stroke. *C* will advance to the left to reduce the work output of the cycle or will move to the right to increase the work of the cycle.

In throttle-governed engines the cutoff *C* stays in a fixed position, but a throttle valve varies the steam pressure entering the cylinder. This raises and lowers the admission line to vary the area of the card, i.e., the work output.

12-23. Basic Engine Types. Figure 12-30 shows the three types of steam engines: (*a*) single-valve, (*b*) four-valve counterflow, and (*c*) the uniflow. In (*a*) steam from a steam chest enters through a form of valve to the steam port leading to the cylinder. The piston moves through one stroke during the admission and expansion phases. During the reverse piston motion the cooled steam exhausts through the same port to an exhaust passage. This cooling of metal causes wasteful condensation during the succeeding stroke.

In (*b*), the four-valve counterflow (Corliss) design, cylinder condensation has been reduced. Four separate valves control steam flow although the steam counterflows in the cylinder on each side of the piston. This design has low clearance volumes because of the short steam ports. The exhaust steam does not pass through the inlet port and cool it. The engine has a more complicated mechanical design to control the valves in proper sequence.

The uniflow engine in (*c*) is the most efficient reciprocating engine built today. Steam enters one end of the cylinder and expands until the piston uncovers a ring of exhaust ports at the center of the cylinder. Most of the steam exhausts by blowing out here, which avoids marked cooling of the cylinder walls. This engine has full stroke compression and is satisfactory for condensing operation. For back-pressure operation built-in clearance pockets prevent excess compression pressures. An auxiliary exhaust valve, mechanically operated, may be used to allow adding more steam per stroke to gain capacity. The auxiliary valve allows additional area to exhaust the steam than can be passed through the normal exhaust ports.

12-24. Steam-engine Performance. Several ratios define the performance of steam engines.

$$\text{Brake horsepower (bhp)} = \frac{2\pi nFd}{33{,}000} \qquad (12\text{-}26)$$

where n = rpm shaft speed
F = net brake force, lb
d = brake arm length, ft

$$\text{Indicated horsepower (ihp)} = \frac{pALN}{33{,}000} \qquad (12\text{-}27)$$

where p = mep, psi
A = piston area, sq in.
N = number of cycles per minute = n
L = length of stroke, ft

$$\text{Indicated or brake engine efficiency} = \frac{2{,}545}{w(h_1 - h_2)_s} \qquad (12\text{-}28)$$

where w = steam rate, lb per ihp-hr or lb per bhp-hr
$(h_1 - h_2)_s$ = isentropic enthalpy drop for given conditions

$$\text{Mechanical efficiency} = \frac{bhp}{ihp} \qquad (12\text{-}29)$$

$$\text{Brake or indicated thermal efficiency} = \frac{2{,}545}{w(h_1 - h_f)} \qquad (12\text{-}30)$$

where h_1 = initial enthalpy at throttle, Btu per lb
h_f = enthalpy of saturated liquid at exhaust pressure, Btu per lb
w = steam rate, lb per bhp-hr or lb per ihp-hr

These performance factors vary with load and usually are plotted on graphs with load as abscissa. The unit steam-rate curves prove the most practical for comparing performance. Figure 12-31 shows typical performance curves for a 1,000-hp uniflow engine with different inlet and exhaust conditions. This engine performs best at about half load. Design could make the engine perform best at full load if so desired.

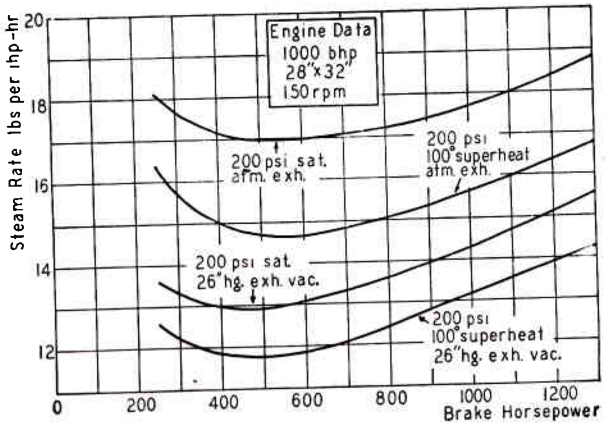


FIG. 12-31. Uniflow-engine steam rates.

Table 12-3 shows average performance levels for engines of several types. The economy depends on the steam conditions, as indicated for the uniflow engine.

TABLE 12-3. RECIPROCATING-ENGINE STEAM RATES

Engine type	Dry-inlet steam pressure, psig	Full-load steam rate, lb per ihp-hr
Simple, high-speed, single-valve, noncondensing	80–100	30–35
Compound, high-speed, single-valve:		
Noncondensing	150–200	24–26
Condensing	150–200	19–22
Four-valve, simple, counterflow:		
Noncondensing	80–100	24–26
Condensing	80–100	20–22
Four-valve, compound, counterflow:		
Noncondensing	150–200	20–22
Condensing	150–200	14–16

PROBLEMS

12-1. A 7,500-kw and a 5,000-kw turbogenerator are installed in a plant operating with steam at 400 psig, 750 F, and a back pressure of 2 in. Hg. Steam data for the units in pounds per hour are as shown in Table P-1.

TABLE P-1

5,000 kw load		7,500 kw load	
½	¾	½	¾
27,800	51,100	41,500	75,000

A system load of 9,500 kw is divided equally between the units.

Calculate:

a. Combined steam rate

b. Average turbine heat rate

12-2. A 6,000-kw straight noncondensing turbogenerator takes steam at the throttle at 400 psia and 600 F. Exhaust is at 10 psig, and the steam rate is 21.5 lb per kwhr. The generator and turbine mechanical losses are 8.0 per cent.

Calculate:

a. Over-all engine efficiency

b. Exhaust-steam conditions

12-3. A nonextraction back-pressure turbine receives steam at 350 psia and 200-F superheat with exhaust at 25 psig.

For a steam consumption of 35.0 lb per kwhr, calculate:

a. Over-all engine efficiency

b. Over-all thermal efficiency

12-4. Steam at 650 psia and 780 F is used in an extraction turbine having five pressure stages. Exhaust pressure is 1.0 in. Hg abs. Stage data are as shown in Table P-2.

TABLE P-2

Stage No.	Pressure, psia	Stage efficiency, per cent
1	200	81.3
2	70	76.9
3	18	74.3
4	3.5	71.5
5	1.0 in. Hg.	68.8

Steam is bled from the third stage for process steam.

a. Plot the condition curve, and determine the condition of the bled steam.

b. Calculate the process steam required in pounds per hour if 30 gpm of water is heated through a 120-F range, in a nonmixing heater. Assume that the condensate leaves the heater subcooled 10 F.

12-5. The data of Table P-3 were obtained from a straight-condensing turbine rated at 1,500 kw with steam at 215 psia and 500 F exhausting to a pressure of 2.5 in. Hg abs.

TABLE P-3

Load, kw	Throttle flow, lb per hr	Load, kw	Throttle flow, lb per hr
400	7,500	1,400	19,700
600	10,000	1,600	22,300
800	12,500	1,800	25,300
1,000	15,000	2,000	28,600
1,200	17,300	2,200	33,000

Plot the following performance curves:

a. Input-output curve based on input in Btu per hour

b. Steam rate

c. Engine efficiency

12-6. The data of Table P-4 were obtained from a 3,000-kw straight-condensing turbine with steam at 400 psia and 700 F exhausting to a pressure of 2 in. Hg abs.

TABLE P-4

Load, kw	Throttle flow, lb per hr	Load, kw	Throttle flow, lb per hr
3,000	31,260	1,500	16,950
2,250	24,075	750	10,125

Plot the following performance curves:

a. Input-output curve as the Willans line

b. Steam rate

c. Heat rate

d. Thermal efficiency

12-7. Calculate the turbine heat rate, over-all thermal efficiency, and over-all engine efficiency for an extraction turbine in a regenerative cycle based on the following data:

Turbogenerator load, 100,000 kw
Throttle pressure, 1,250 psia
Throttle temperature, 900 F
Exhaust pressure, 1.0 in. Hg abs
Throttle flow, 915,000 lb per hr
Extraction conditions as in Table P-5

TABLE P-5

Pressure, psia	Extraction, lb per hr	Pressure, psia	Extraction, lb per hr
645	33,000	103	69,750
522	65,100	22.6	42,100
254	51,450	6.9	53,100

Feedwater temperature leaving last heater, 485 F

12-8. Determine the turbine heat rate and over-all engine efficiency for an extraction turbine in a regenerative cycle based on the following data:

Turbogenerator load, 10,000 kw
Throttle pressure, 380 psia
Throttle temperature, 950 F
Exhaust pressure, 1.0 in. Hg abs
Throttle flow, 84,450 lb per hr
Extraction conditions, 6,220 lb per hr at 125 psia, 5,420 lb per hr at 40.0 psia, 6,760 lb per hr at 10 psia
Feedwater temperature leaving last heater, 340 F

12-9. An industrial power plant must supply 100,000 lb per hr of dry saturated steam at 5.3 psig from the turbine exhaust. A boiler and turbogenerator are to be installed to meet an electric load of 5,500 kw. An over-all electrical and mechanical efficiency of 90.0 per cent and an over-all engine efficiency of 67.5 per cent are expected. Calculate the boiler pressure and temperature to be used, neglecting line pressure drops.

12-10. Steam at 1,250 psia and 950 F drops 120 psi in pressure through an impulse stage. The nozzle makes a 15° angle with the blade entrance and is 92 per cent efficient. The blade is symmetrical, and blade friction reduces the relative velocity by 12 per cent. Blade velocity ratio is 0.45. Determine the steam work against the blade face per pound of steam flow per second.

12-11. A nozzle makes an angle of 17° with the blade direction of motion and discharges steam at 2,200 ft per sec. Blades 6 in. long are fastened to the disk, whose diameter is 3.5 ft. Blades are symmetrical, with 18° entrance angles. Compute the wheel speed (rpm) for smooth (shockless) entrance of steam on the blades.

12-12. A nozzle expands steam from 2,400 psia and 1100 F through a pressure drop of 250 psi. Compute the jet power for a flow of 750,000 lb per hr with a nozzle efficiency of 95 per cent.

12-13. A reaction stage receives steam at 600 psia and 580 F and expands the steam through a pressure range of 50 psi. A flow of 30,000 lb per hr occurs, and at this rate the stage has an efficiency of 76 per cent. Compute the blade work done by the steam.

12-14. Compute the heat rates of the 66-mw and the 100-mw AIEE-ASME standard units. State carefully the basis for each computation.

12-15. Steam with a velocity of 2,000 ft per sec passes over a set of blades at the rate of 58,000 lb per hr. The nozzle angle is 15° at entrance, the blade entrance angle is 28°, the exit angle is 20°, and a relative velocity loss of 10 per cent occurs in the blade.

Compute:

a. Blade power

b. Blade efficiency

c. Blade velocity ratio

12-16. A mechanical-drive turbine has one set of nozzles and blades. Available energy to the nozzle is 120 Btu per lb, and 2,200 lb per hr flows. The blades are symmetrical and have a blade velocity ratio of 0.45. The nozzle efficiency is 96 per cent, blade friction reduces relative velocity by 8 per cent, and the blade entrance angle is 22°. Draw a vector diagram as in Fig. 12-6*a*, and compute the shaft horsepower based on a mechanical efficiency of 96 per cent.

12-17. A 10- by 12-in. double-acting uniflow engine has a 1.8-in. piston rod and no tail rod. Indicator-card data are as follows:

Head-end card, 1.32 sq in.
Crank-end card, 1.44 sq in.
Card length, 2.90 in.
Spring scale, 80 psi per in.
Brake arm, 60 in. long
Net brake weight, 172 lb
Speed, 250 rpm

Calculate:
a. Indicated horsepower
b. Brake horsepower
c. Friction horsepower
d. Mechanical efficiency

12-18. A steam engine develops 60 bhp with dry saturated steam at 150 psia and exhaust at 18.0 psia. Steam consumption is 1,620 lb per hr.

Calculate:
a. Brake thermal efficiency
b. Indicated engine efficiency based on 95 per cent mechanical efficiency
c. Thermal efficiency of equivalent Rankine engine

12-19. An 8- by 10-in. double-acting engine has a clearance volume of 8 per cent. Estimate the indicated horsepower if expansion and compression follow the law $pV^{1.00} = C$, based on the following data for a single-valve type.

Cutoff, 40 per cent
Average speed, 320 rpm
Steam inlet pressure, 150 psig
Exhaust pressure, 3 psig
Terminal compression pressure = inlet pressure

12-20. A 9- by 10-in. double-acting engine operating at 320 rpm develops 45.0 ihp. Calculate the diagram factor from the following:

Steam inlet pressure, 100 psig
Exhaust pressure, 2 psig
Cutoff, 30 per cent
Clearance, 8 per cent

12-21. A 10- by 12-in. steam engine with steam at 100 psia has a connecting rod 30 in. long. Calculate the crankshaft torque when the piston has moved 1.75 in. from the beginning of its stroke. The engine is single-acting.

12-22. A uniflow engine has a steam consumption of 17.0 lb per ihp-hr. Operation is under the following conditions:

Steam pressure, 200 psia
Steam temperature, 450 F
Back pressure, 2.3 psig

Calculate:
a. Indicated engine efficiency
b. Indicated thermal efficiency
c. Exhaust-steam enthalpy and quality, assuming radiation losses 3.0 per cent of indicated power
d. Available process heat in Btu per hour if, at above rate, 500 bhp is developed at 96 per cent mechanical efficiency and condensate returns at 150 F

12-23. A steam engine operates under the following conditions:

Steam pressure, 200 psia
Steam temperature, 400 F.
Condenser pressure, 4 in. Hg
Engine efficiency, 52 per cent

Calculate:
a. Actual steam rate
b. Steam cost per year with steam at 40 cents per 1,000 pounds

12-24. A steam-driven pumping engine delivers 28×10^6 gal per day of water. A pump discharge gauge shows 120 psig and is attached 10 ft above the center line, while the gauge on the suction side reads 15.1 in. Hg vacuum and is attached 3 ft below the center line. The pump duty is 97×10^6 ft-lb per million Btu based on pump work and $h_g - h_f$ of steam. Steam pressure is 200 psia at 500 F. Exhaust pressure is 3.0 psia.

Calculate:
a. Pump horsepower
b. Steam flow in pounds per hour

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CHAPTER 13

CONDENSERS

13-1. Condenser Theory. Power-plant cycles improve in efficiency as the turbine exhaust pressure drops. By condensing the steam exhausted from a turbine the condensers maintain a low pressure at the exhaust. Short connections between turbine exhaust and turbine inlet keep pressure drops low. Connections vary with size of equipment and design, from solid welded joints to water-sealed flexible joints.

Condensers resting directly on foundations connect to the turbine with an expansion joint, but condensers standing on spring supports may be connected solidly.

Most power plants stand on the shore of a river, lake, bay, or ocean so that they have access to large volumes of cooling water. The water acts as the heat-removal agent in the condensers. Figure 13-1 shows schematically the two basic types: (1) surface and (2) direct-contact, or jet. Uncontaminated condensate can be recovered in surface condensers, since the cooling water does not mix with the condensing steam. Direct-contact condensers mix the steam with the cooling water and are infrequently used in power plants. Steam pressure in a condenser depends mainly on the flow rate and temperature of the cooling water and on the effectiveness of air-removal equipment.

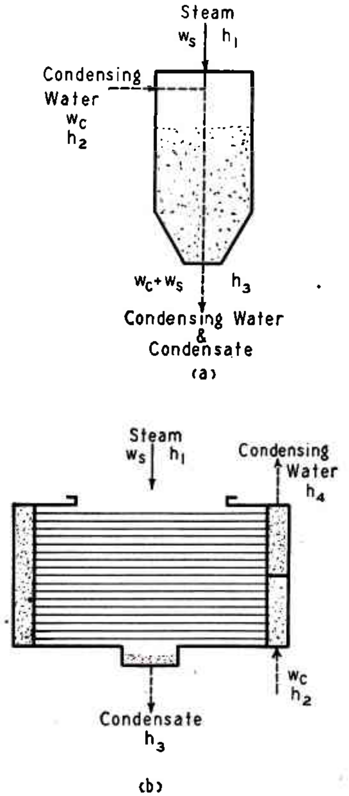

FIG. 13-1. Diagrammatic representation. (*a*) Direct-contact condenser; (*b*) surface condenser.

Energy exchange in a condenser may be analyzed by means of the steady-flow equation (Fig. 13-1):

Direct: $$w_s h_1 + w_c h_2 = (w_s + w_c) h_3 \qquad (13\text{-}1)$$

Surface: $$w_s(h_1 - h_3) = w_c(h_4 - h_2) \qquad (13\text{-}2)$$

where w_s = exhaust-steam flow rate, lb per hr
w_c = cooling-water flow rate, lb per hr
h = enthalpy, Btu per lb (Fig. 13-1)

The exhaust-steam enthalpy may be found from the turbine condition line corrected for exhaust loss or by the energy relation

$$h_1 = \frac{h_i - W - Q - \Sigma mh}{1 - \Sigma m} \qquad (13\text{-}3)$$

where h_1 = exhaust-steam enthalpy, Btu per lb
h_i = prime-mover inlet-steam enthalpy, Btu per lb
W = work output to turbine blades, Btu per lb steam
Q = radiation or other heat loss, Btu per lb inlet steam
Σmh = enthalpy of turbine extraction steam, Btu per lb inlet steam
Σm = total extracted steam, lb per lb entering steam

In a surface condenser:

Terminal temperature difference = (steam temperature) − (cooling-water exit temperature)

This is usually 5 F or more. A low cooling-water temperature rise helps keep condensing-steam pressures low.

13-2. Surface Condensers. These units, shell-and-tube heat exchangers, remove the unavailable energy and available energy degraded by irreversibilities from the plant cycle. Practically they maintain low exhaust pressures and recover the condensate to be recirculated as working fluid in the cycle.

Figure 13-2 shows one form of condenser design and names the various parts. The surface condenser consists of a casing or shell with a chamber at each end, called "water boxes." Tube sheets separate the water boxes from the center steam space. Banks of tubes connect the water boxes by piercing the tube sheets; the tubes essentially fill the shell or steam space. Circulating pumps force the cooling (circulating) water through the water boxes and the connecting tubes.

Tubing and water boxes may be arranged for *single-pass* or *two-pass* flow of water through the shell. In single-pass (Fig. 13-2) water enters the right water box and flows once through all the tubes in parallel and leaves through the left water box. In two-pass (Fig. 13-3) water flows through the bottom half of the tubes in one direction, reverses in the far water box, and returns through the upper tubes to the near water box. Water enters and leaves through the same water box, which is divided into two chambers by a horizontal plate.

A large cooling-water supply allows using single-pass units. The water-temperature rise can be less than for two-pass, with less surface area since the logarithmic mean temperature difference (LMTD) increases. Limited cooling-water supplies leads to using two-pass condensers. Using more than two passes raises water-pumping costs to uneconomical levels in almost all cases.

Most modern condensers have *divided water boxes*, vertical partitions giving two parallel flow paths through the shell. This permits taking half the condensing surface out of service for cleaning while water flows through the other half to keep the unit running at reduced load.

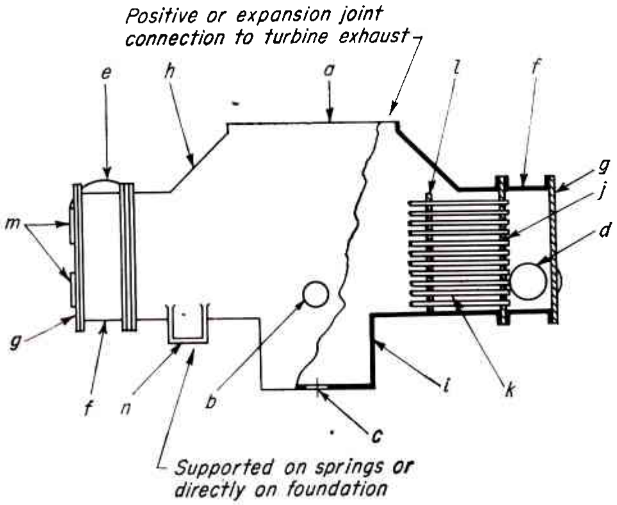
Positive or expansion joint connection to turbine exhaust
e
h
a
l
f
g
j
d
m
g
f
n
b
c
i
k
Supported on springs or directly on foundation

FIG. 13-2. Diagram of surface-condenser elements to establish terminology: *a*, Steam inlet; *b*, air-vapor outlet; *c*, condensate outlet; *d*, circulating (cold-water) inlet; *e*, circulating (discharge-water) outlet; *f*, water box; *g*, water-box cover; *h*, condenser shell; *i*, hotwell; *j*, tube sheet; *k*, tubes; *l*, tube-support plate; *m*, handholes or manholes; *n*, condenser-support bracket. (*Reprinted with modifications from "Standards for Steam Surface Condensers," 4th ed. Copyright* 1955 *by the Heat Exchange Institute,* 122 *East* 42*nd St., New York* 17, *New York.*)

Figure 13-3 shows some water-box nozzle arrangements. Condenser design must account for the moments produced on the unit by the difference in water pressure at the inlet and discharge connections. The shell and supports must be strong enough to support the condenser when completely filled with water in the steam space. This is done to check the tube tightness in the tube sheets. Handholes in the water-box covers give access to tubes for cleaning and replacement.

Water boxes are often of cast iron, although bronze or fabricated steel with Neoprene and Koroseal linings have also been used. Newer units have fabricated-steel shells, while older ones have been cast. Cross sections of all shapes may be found in condensers, though circular and square are mostly used. Larger units over 50,000 sq ft usually have a rectangular or oval shape to get more surface in an area with limited headroom.

The hotwell stores the condensate and keeps a head on the condensate pump. One or more air offtakes in the steam space lead accumulating air to the air-removal pump. The air offtakes carry a mixture of air-rich vapor, most of the vapor being recovered by condensation.

13-3. Condenser Tubes. The tubes that provide the heat-transfer surface in the condenser are fastened into tube sheets, usually made of Muntz metal. Older designs have the tubes rolled into one tube sheet and fastened into the other with some sort of packing or ferrule, to allow for thermal expansion. Many modern designs have tubes rolled into both tube sheets, and for ultratightness tubes are welded into the tube sheet. An expansion joint in the shell takes care of expansion in these designs.

Support sheets, usually steel, prevent the tubes from sagging between tube sheets; they also help to partition steam for uniform distribution in the shell.

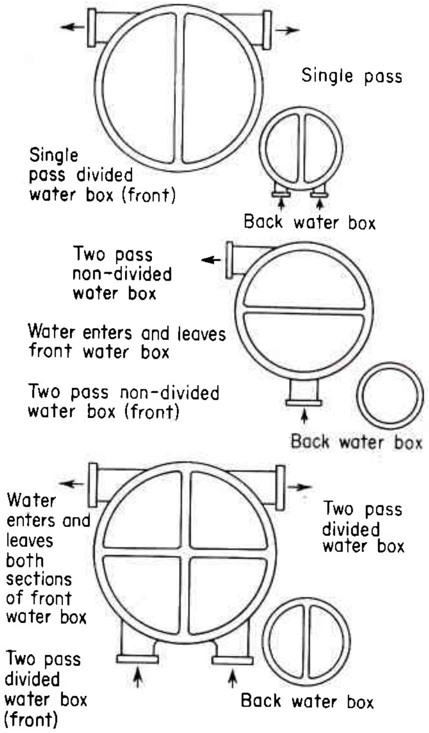
Single pass
Single pass divided water box (front)
Back water box
Two pass non-divided water box
Water enters and leaves front water box
Two pass non-divided water box (front)
Back water box
Water enters and leaves both sections of front water box
Two pass divided water box
Two pass divided water box (front)
Back water box

FIG. 13-3. Typical arrangement of water-box nozzles with *single-* and *double-*pass and *divided* and *nondivided* design. (*Courtesy of Allis-Chalmers Manufacturing Company.*)

Some typical condenser-tube materials include:

Cupronickel 30%—70% *Cu*, 30% *Ni*, 0.5% *Fe*
Aluminum bronze 5%—95% *Cu*, 5% *Al*
Aluminum brass—76% *Cu*, 22% *Zn*, 2% *Al*, 0.05% *Ar*
Admiralty—71% *Cu*, 28% *Zn*, 1% *Sn*, 0.05% *Ar*
Muntz—60% *Cu*, 40% *Zn*

Condenser-tube water flows range from about 5 to 10 ft per sec. Higher flows raise pumping-power need and erode tubes at their entrances, shortening their life considerably. Tubes are installed with an upwardly bowed arc. This aids drainage in a shutdown condenser and helps prevent tube vibration.

As many as four low-pressure feedwater heaters have been installed in the condenser steam-inlet section (neck) in some recent units. This can be done when pressure losses are low; the advantages are saving in machinery space and shortening the bleed pipe to the turbine. The latter reduces pressure losses in bleed steam, keeping cycle efficiency high.

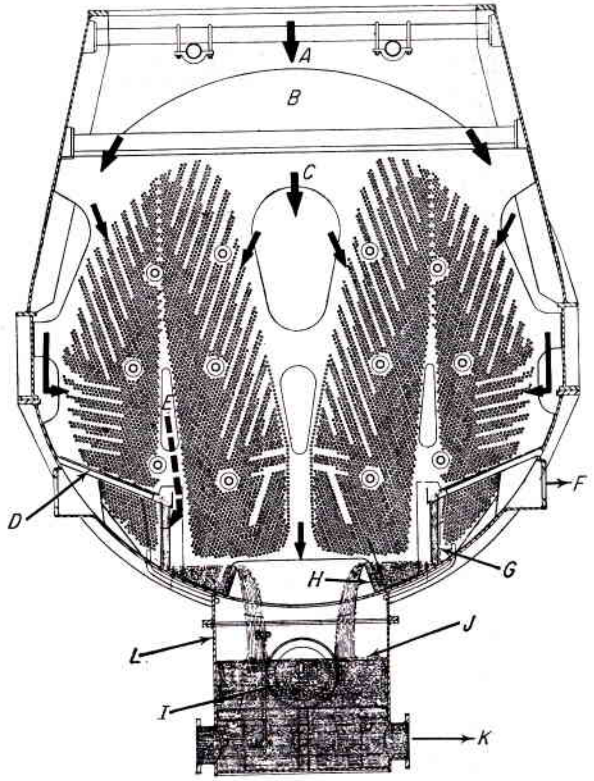

Fig. 13-4. Cross section of typical, large surface condenser. Note *steam lanes* to improve flow conditions. *A*, steam inlet; *B*, support-plate cutouts; *C*, central steam lane; *D*, insulating baffles; *E*, air-exit lane; *F*, air offtake; *G*, drain pipes; *H*, condensate weirs for condensate reheating and deaeration; *I*, manhole; *J*, strainer; *K*, condensate pump suction; *L*, hotwell. (*Courtesy of Worthington Pump and Machinery Corporation.*)

13-4. Tube-sheet Layout. The arrangement of tubes in the steam space controls the effectiveness of the condenser. Older designs tended to crowd a maximum number of tubes inside the shell to make as much surface available as possible. Modern designs have steam lanes between tube banks to get maximum steam flow with least pressure drop and uniform distribution in the shell and to maintain constant steam velocities over the tubes.

Baffles in the steam space direct steam flow for even distribution and keep steam away from the air-offtake zones (Fig. 13-4). Air offtake removes the air-vapor mixture by making it pass over the coldest tubes to wring out much of the vapor by condensation. This reduces the air-pump load.

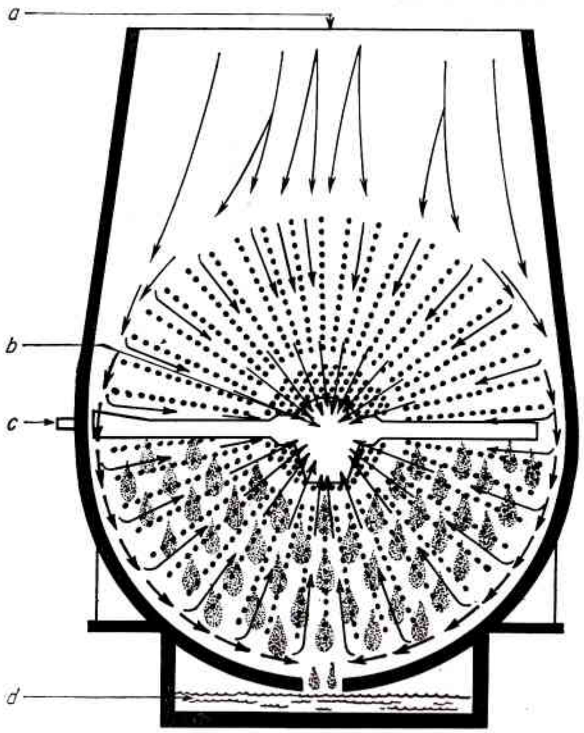

Fig. 13-5. Steam-flow paths and air-removal sections of a condenser using the radial-flow principle: *a*, Steam inlet; *b*, air cooler; *c*, air-removal section; *d*, condensate. (*Courtesy of Westinghouse Electric Corporation.*)

Weirs cascade the condensate into sheets into a deaerating hotwell through an atmosphere of steam. This reheats the condensate if it has been subcooled and boils out any dissolved gases.

Figure 13-5 shows the tube sheet of a radial-flow condenser. Steam flows round the tube bundle and then in short radial paths toward the center air offtake. The air-cooler section surrounds an open passage connected to the air offtake at the cold end of the condenser.

Figure 13-6 shows the internal construction of a large divided-water-box two-pass surface condenser. Air-offtake passages stand at the center of the steam space. Tube-support sheets hold tubes not shown, the lines indicating tube-hole centers.

13-5. Direct-contact Condensers. These units mix the cooling water with the steam to condense it. They are less expensive, can use fairly dirty cooling water and need less of it, and require little maintenance. The complete loss of condensate offsets these advantages, however. The incoming cooling water (Fig. 13-7) carries dissolved gases into the steam space to be liberated there. This reduces vacuum and loads up air-removal equipment.

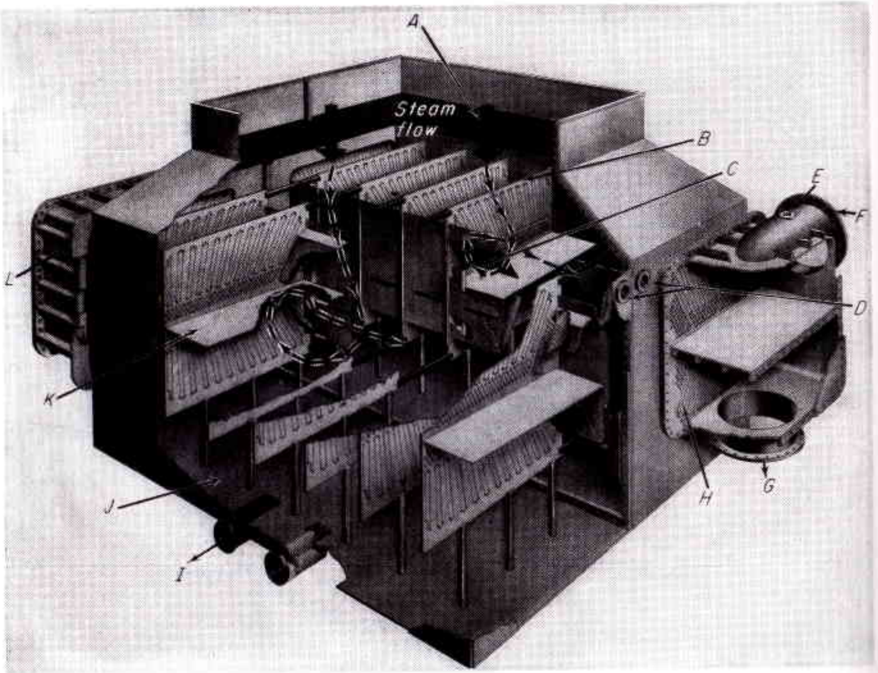

FIG. 13-6. Phantom view of typical surface condenser of *two-pass divided-water-box* design. *A*, exhaust from turbine; *B*, tube-support plate; *C*, air cooler; *D*, air offtake; *E*; vent; *F*, circulating-water inlet; *G*, circulating-water outlet; *H*, tube sheet; *I*, to condensate pump; *J*, hotwell; *K*, condensate drain tray; *L*, shell expansion joint. (*Courtesy of Foster Wheeler Corporation.*)

These condensers have three forms: (1) barometric, (2) low-level jet, and (3) ejector jet. The barometric condenser (Fig. 13-7) uses the gravity head of the mixed condensate and cooling water to discharge it to waste. Supply water coming in above the condenser makes pumps unnecessary.

A low-level jet condenser can be placed directly under a turbine. It has the same mixing section as the barometric unit, but a pump replaces the long pipe needed for gravity discharge.

The ejector-jet condenser also fits under a turbine; it has high-velocity water jets to condense incoming steam within a mixing head. The fast-

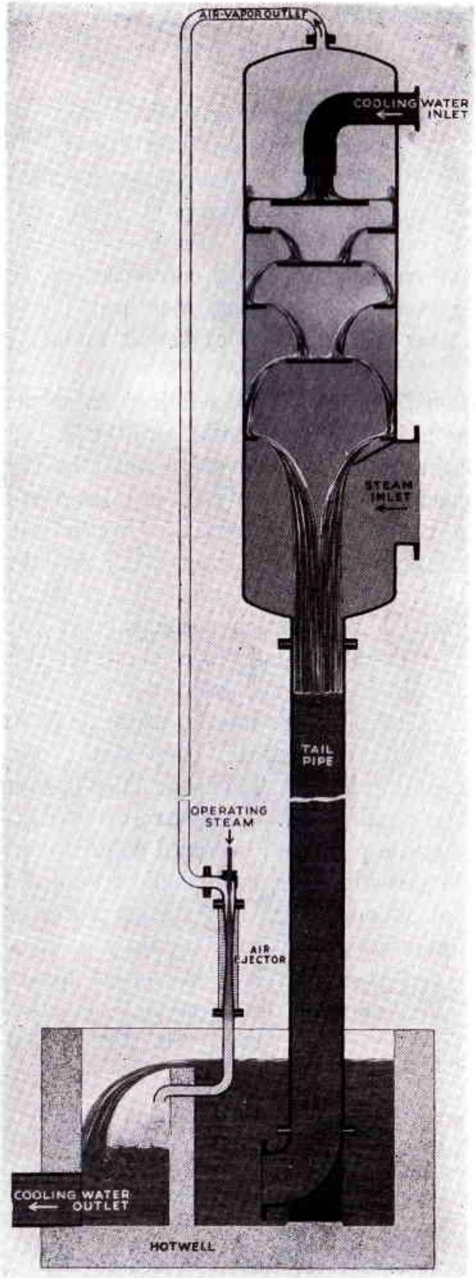

FIG. 13-7. Typical barometric condenser. Note that tail pipe must be long enough to contain an atmospheric head of water (34± ft). (*Courtesy of Ingersoll-Rand Company.*)

moving mixture passes through a diffusing head to atmosphere. Water must be pumped to the condenser jets, but the diffuser removes the mixture to maintain vacuum.

The three types have mixing heads performing the same function, but they use different methods of removing the water mixture.

13-6. Condenser Auxiliaries. A condenser needs equipment to move cooling water through the tubes, remove air from the steam space, and drain condensate from the hotwell. Such equipment includes: (1) circulating- or cooling-water pumps; (2) condensate or hotwell pumps; (3) air-removal equipment; (4) priming ejectors; (5) atmospheric relief valve; (6) inlet-water tunnel, pipe, or canal; and (7) discharge tunnel, pipe, or canal.

A condenser uses 75 to 100 lb of cooling water per pound of steam condensed. Large units need tremendous water flows; to keep the pump work to a minimum, condenser elevation must be close to water level. The pump then needs to develop only enough head to overcome the flow resistance of the cooling-water circuit. Circulating pumps for condensers are of three types: (1) centrifugal, (2) mixed flow, and (3) propeller (see Sec. 15-2).

Condensate pumps handle much smaller flows than the circulating pumps. They must develop heads to push water through atmospheric pressure, pipe friction, heater water-circuit friction, and the elevation of the surge tank. These pumps handle water at low absolute pressure; to prevent flashing the condensate, they stand below the hotwell to receive a positive suction head. Their specially designed glands prevent air leakage into the condensate. They are often installed in duplicate, and oversized to handle possible abnormal conditions.

Noncondensable gases like air, carbon dioxide, and hydrogen get into the steam space of a condenser. They may enter through leakage at glands, valve bonnets, or porous walls or may be in the throttle steam. Those gases not dissolved by the condensate diffuse throughout the steam space of the condenser. As these gases accumulate, their partial pressure raises the condenser total pressure. The total condenser pressure is

$$P_c = P_s + P_a \qquad (13\text{-}4)$$

where P_s = steam saturation pressure corresponding to steam temperature

P_a = air pressure (moisture-free)

This equation shows that a low-temperature condensing water is needed to develop a low back pressure P_s. Air must be removed constantly to keep P_a low. In removing this air it will always have some entrained vapor; the mixture must be compressed and discharged to atmosphere.

Though the mass of air may be low, its low pressure shows that it occupies a very large volume, which must be handled by the air-removal equipment. The air offtakes withdraw the air-vapor mixture from the steam space over a cold section of the condenser tubes (Fig. 13-8) or through an external cooler. This condenses part of the moisture and increases the air-to-steam ratio. Steam jets or mechanical vacuum pumps compress the air to atmospheric.

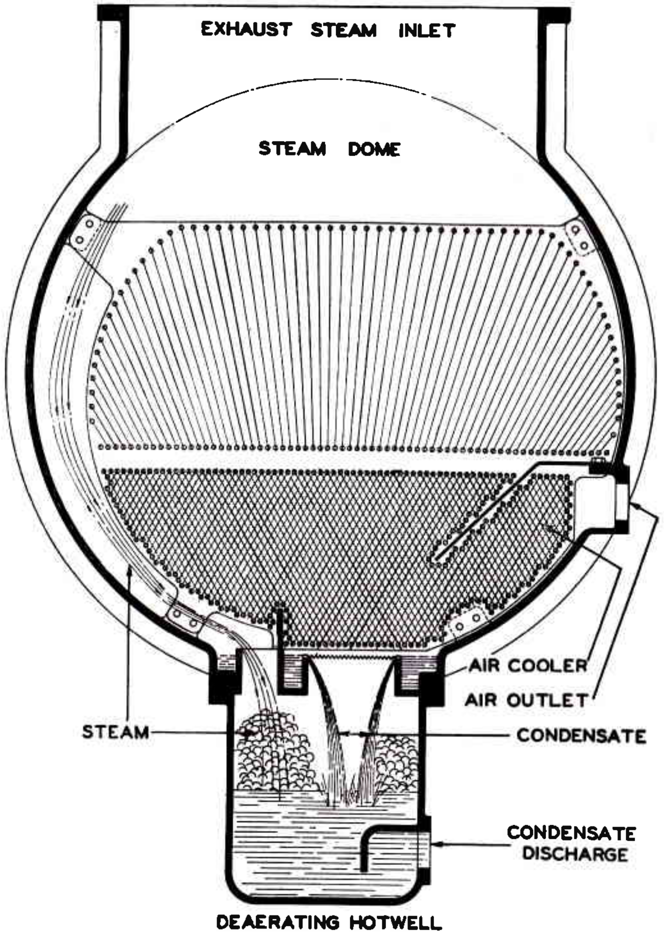

FIG. 13-8. Condenser cross section illustrating air-removal section across air-cooler tubes. (*Courtesy of Elliott Company.*)

Figure 13-9 shows a two-stage air ejector. Medium-pressure steam expands through a set of nozzles at high velocity. The steam jet entrains the air-vapor mixture from the air offtake and discharges to a higher pressure through a diffuser section. The high vacuum in a condenser makes it economical to remove the air in two compression stages. The two stages run in series with a small intercondenser between them to condense part of the vapor and jet steam in the mixture. The jet of the second-stage ejector picks up the saturated air leaving the intercondenser and pressurizes it to atmospheric. The mixture flows through an aftercondenser to remove the condensable vapor and discharges saturated air to atmosphere.

Condensers normally use steam-jet air ejectors with twin elements, one acting as a spare. The inter- and aftercondensers use main condensate to recover the heat. Condensate from the inter- and aftercondensers is trapped to the main condenser hotwell; Fig. 13-10 shows a twin-element unit.

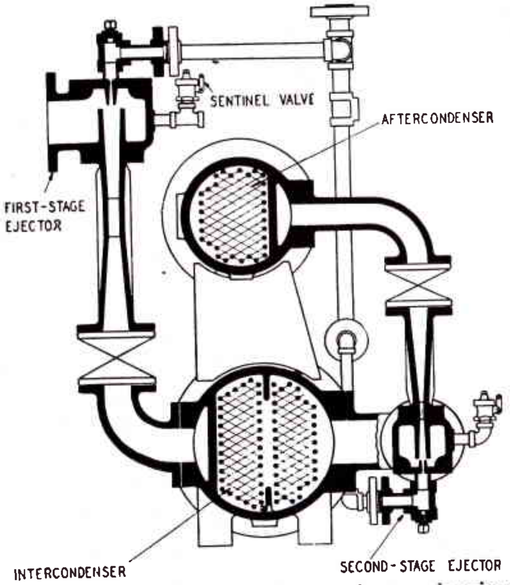

Fig. 13-9. Cross section of two-stage steam-jet air ejector, showing principle of operation. Main condensate used as cooling water for intercondenser and aftercondenser. (*Courtesy of Elliott Company.*)

During starting, large condensers use "hogging jets" to supplement the air ejectors and pull a vacuum in a shorter time.

Steam jet ejectors are simple, reliable, and low in cost. They work at low pumping efficiency, and the steam for the jets must be throttled in high-pressure plants. This severely erodes the throttling valves, entailing high maintenance. To avoid this, mechanical vacuum pumps (Fig. 13-11) have been applied in some high-pressure plants. They remove air during normal operation and start-up. A viscous fluid, tricresyl phosphate, maintains an airtight seal between rotor and casing. In condenser service a whirl-type separating chamber on the discharge side separates air and moisture.

13-7. Condenser Performance and Installation. Figure 13-12 shows the important operating characteristic of a surface condenser. Each condenser can develop a certain vacuum or back pressure for given inlet cooling-water temperature and steam flow. These curves apply to a given cooling-water flow through the tubes. Another similar graph would be needed to show these relations with a different flow.

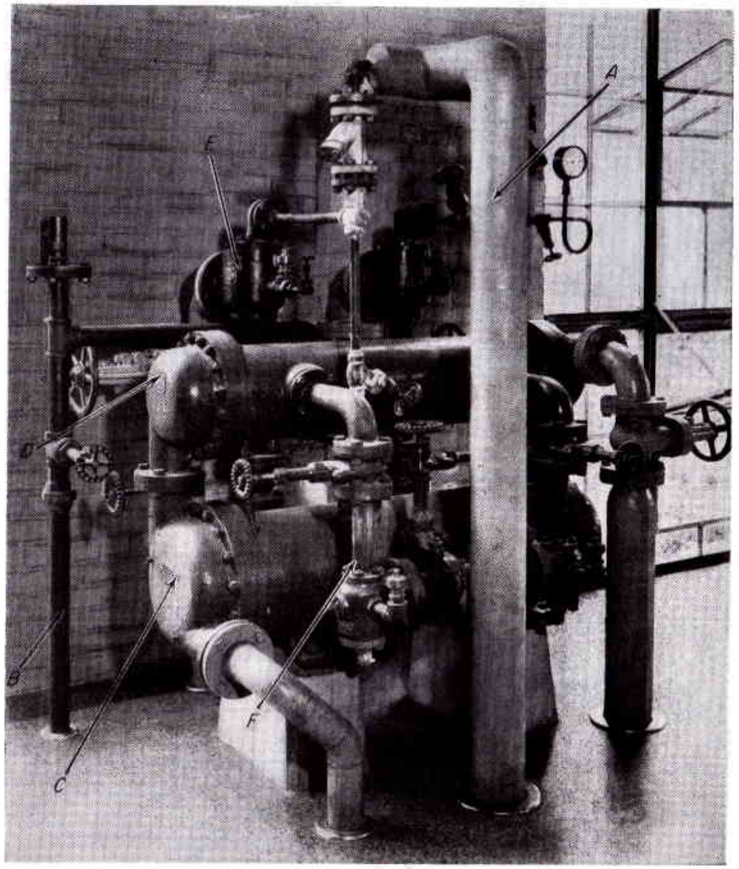

Fig. 13-10. Two-stage steam-jet air-ejector installation. *A*, Supply steam; *B*, air-vent line; *C*, intercondenser; *D*, aftercondenser; *E*, first-stage jets; *F*, second-stage jets. (*Courtesy of Elliott Company.*)

The vacuum drops (or back pressure rises) with increasing steam flow at a constant cooling-water temperature and flow. Small changes in back pressure markedly affect the turbine heat rate, and so plant sites

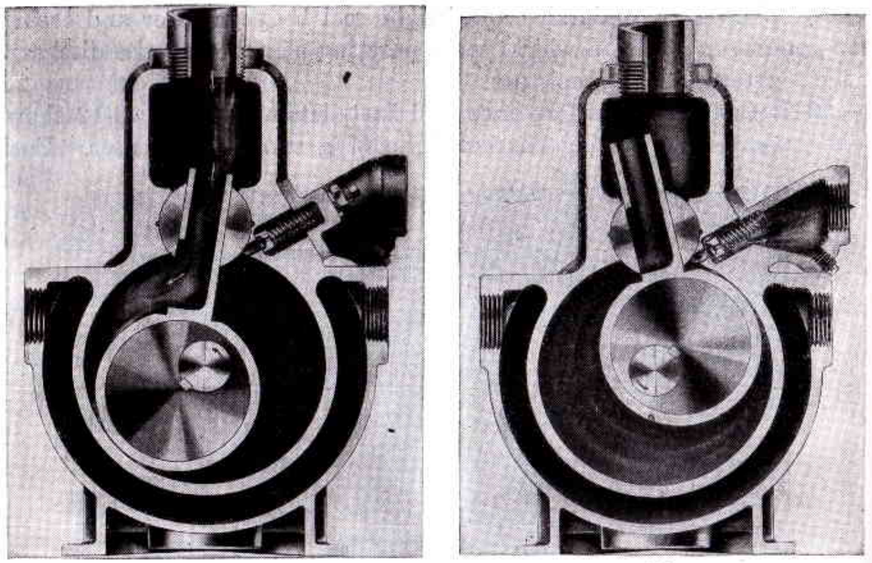
(a) (b)

FIG. 13-11. Rotary air-removal pump has eccentric rotor that creates space for gas entering suction at top, while compressing previous charge to discharge to a separator. *(Courtesy of Kinney Manufacturing Division, The New York Air Brake Company.)*

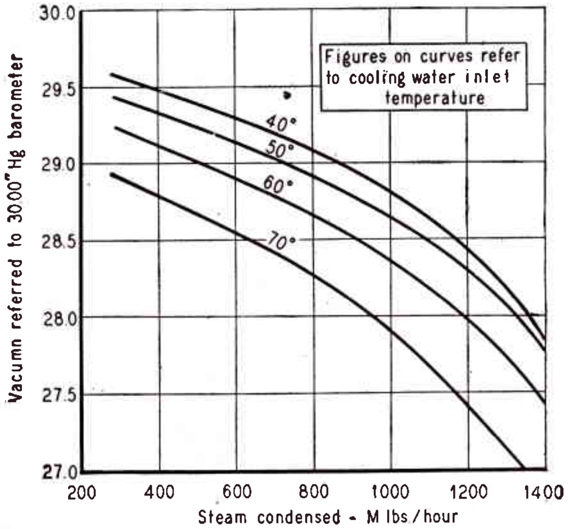

FIG. 13-12. Performance characteristics of typical surface condenser. Curves are based on given circulating-water flow rate at all temperatures.

must be chosen to find the most favorable cooling-water conditions. Figure 13-13 shows a typical surface condenser and its auxiliaries.

Condenser performance depends greatly on the cleanliness of the tube heat-transfer surfaces. Cooling water carries in debris and algae that foul up the tube interiors and reduce heat-transfer rates. Tubes may be cleaned by shooting plugs and brushes through them with compressed air. A new method periodically injects slightly oversized rubber balls into the cooling-water inlet, so that the water carries the balls through the tubes to rub them clean. A separator removes the balls automatically from the water at the condenser outlet. Another method reverses the water flow through the condenser tubes at intervals to flush debris off the tube sheets. A reverse-flow valve may be used to do this, or more recently reverse-flow pumps have been made available.

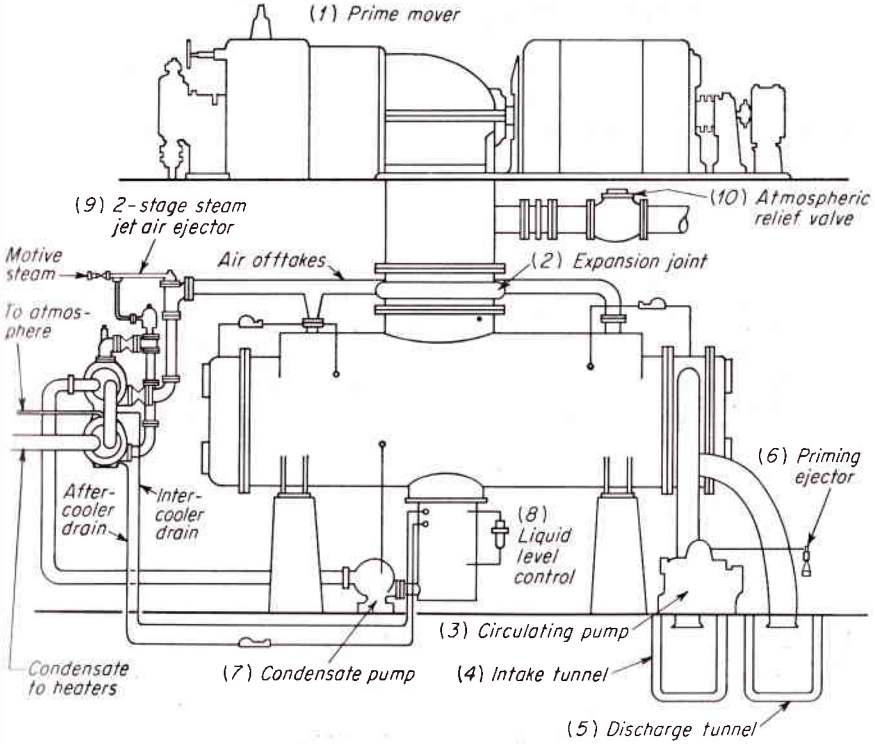

FIG. 13-13. Typical surface-condenser installation showing relation of auxiliaries to condenser. (1) Prime mover; (2) expansion joint; (3) circulating pump; (4) intake tunnel; (5) discharge tunnel; (6) priming ejector used to fill pump when starting; (7) condensate pump; (8) liquid-level control; (9) steam-jet air ejector; (10) atmospheric relief valve. Drains and traps are essential accessories, as also are meters and gauges. *(Courtesy of Allis-Chalmers Manufacturing Company.)*

If cooling-water flow should be accidentally interrupted, a relief valve must release exhaust steam to atmosphere to prevent building up a destructive pressure in the condenser shell. This may be a vertical-lift valve that starts to open at pressures slightly above atmospheric and is fully open at 5 to 10 psig. Another scheme uses rupture disks in the turbine exhaust hood. These fail at specified pressures and release exhaust steam into the turbine room.

PROBLEMS

13-1. Determine the air-to-steam ratio in a condenser for the following conditions:

$$p_c = 2 \text{ in. Hg abs}$$
$$t_s = 90 \text{ F}$$

13-2. Calculate the condenser pressure based on an air-to-steam ratio of 0.088 and a condenser temperature of 92 F.

13-3. Estimate the pressure in a condenser which receives 25,000 lb of steam of 89.0 per cent quality per hour. Steam temperature based on condensing water is expected to be 92 F, and air leakage is 18 cu ft of free air (14.7 psia and 60 F).

13-4. A jet condenser discharges water at 98 F with inlet water at 46 F. Condenser pressure is 3 in. Hg abs. Calculate the pounds of condensing water per pound of exhaust steam with a quality of 88 per cent.

13-5. The pressure in a surface condenser is 1.5 in. Hg abs. Condensing water enters at 50 F and leaves with a terminal temperature difference of 10 F compared with the steam. The condensate is subcooled 5 F. A turbine of 2,000 kw capacity with a steam rate of 14 lb per kwhr exhausts 85 per cent steam to the condenser.

Calculate:

a. Pounds of condensing water per pound of steam

b. Gallons per minute of condensing water

13-6. A 10,000-kw turbine uses 12.0 lb of steam per kilowatthour at rated load. Steam conditions are 600 psia and 750 F with exhaust at 1.0 in. Hg abs. Condensate is subcooled 5 F, and turbine losses to friction and heat transfer amount to 2 per cent of turbine output.

Calculate:

a. Enthalpy of exhaust steam in Btu per pound

b. Gallons per minute of condensing water based on 10-F rise

13-7. Estimate the steam flow to the steam jets of a one-stage air-removal system in pounds per hour and as a percentage of steam flow to the turbine from the following data:

Turbine rating, 25,000 kw
Water rate, 9.5 lb per kwhr
Initial steam to the jets dry-saturated at 400 psia
Nozzle efficiency, 95 per cent
Efficiency of steam jet compression, 4 per cent
Condenser pressure, 2 in. Hg abs
Air-outlet temperature, 70 F
Air flow, 30 lb of *dry* air per hour from condenser
Exponent for compression of vapor-air mixture, 1.45

13-8. A two-pass divided-water-box condenser has 45,700 sq ft of heat-transfer surface consisting of 1-in.-OD 18 Bwg Admiralty-metal tubes 28 ft long. Steam-turbine exhaust at 1 in. Hg has a flow of 480,000 lb per hr with an enthalpy of 985 Btu per lb, while condensate leaves at 77 F. Circulating water entering at 52 F undergoes a 14-F rise. Calculate the over-all heat-transfer coefficient.

13-9. Discuss the effect of summer operation of the condenser in Prob. 13-8. Based on an assumed inlet circulating-water temperature and other reasonable assumptions, approximate the condenser pressure for this condition.

13-10. A small nonextracting 10,000-kw turbine has a guaranteed steam rate of 12.4 lb per kwhr when exhausting at 1 to 5 in. Hg abs. Cooling water is available at 65 F, and for design purposes 1½-in.-OD 16 Bwg brass tubes are specified, with a 15 F rise in cooling water, which circulates at 5 ft per sec. Steam is at 92 per cent quality on entering the condenser, and it is presumed that fouling will decrease the over-all coefficient by 30 per cent during operation. Inside and outside film coefficients are 960 and 1474 Btu per hr-sq ft-°F, respectively, and k of the tubes is 62 Btu per hr-sq ft-°F-ft. Design a condenser for this turbine, considering tube and boundary resistances as flat surfaces and neglecting any crossflow factor.

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CHAPTER 14

FEEDWATER HEATERS AND EVAPORATORS

14-1. Feedwater Heating. In most power plants the main condensate returns to the steam generator as feedwater. Some make-up may be added to replace losses in the cycle. In a few plants the boiler feedwater may be 100 per cent make-up; in this case the plant turbines exhaust at back pressures above atmospheric to supply steam for other purposes. Feedwater is heated by bleeding steam to heaters as in a regenerative cycle from the main turbine, or by using exhaust steam from auxiliary-drive turbines, or by by-product steam from processes.

Feedwater heating with steam at a lower pressure than boiler pressure usually raises over-all plant efficiency. Early plants had a simple *open heater* which received steam from back-pressure turbines or engines driving plant auxiliaries and which heated the main condensate to 212 F at 14.7 psia. Smaller industrial plants still use this scheme.

Modern central stations have as many as ten heaters bleeding steam at successively lower pressures from the main turbine. The condensate and feedwater flow through these to be heated in steps to feedwater temperature.

Feedwater heaters may be classified as: (1) open, direct-contact, or mixing heaters (simple or deaerating) or (2) closed, noncontact, or surface heaters. Heated feedwater enables steam generators to produce more pounds of steam and avoids severe thermal stressing by cold water entering a hot drum. Preheating feedwater also causes scale-forming dissolved salts to precipitate outside the boiler and removes dissolved oxygen and carbon dioxide, which corrode boiler metal.

14-2. Direct-contact, or Mixing, Deaerating Heaters. In a *deaerating heater* entering water spills over trays or sprays through nozzles to make it present a maximum surface, for heat transfer, to the atmosphere of steam filling the heater. This also removes all dissolved O_2 in excess of 0.03 ml per l from the water. A unit that reduces the O_2 to 0.005 ml is a *deaerator*.

Adding sodium sulfite (Na_2SO_3) to the feedwater or boiler water reduces any remaining dissolved O_2 to a tolerable minimum. Rising temperatures reduce the solubility of gases in water; at boiling, gas solubility approaches zero.

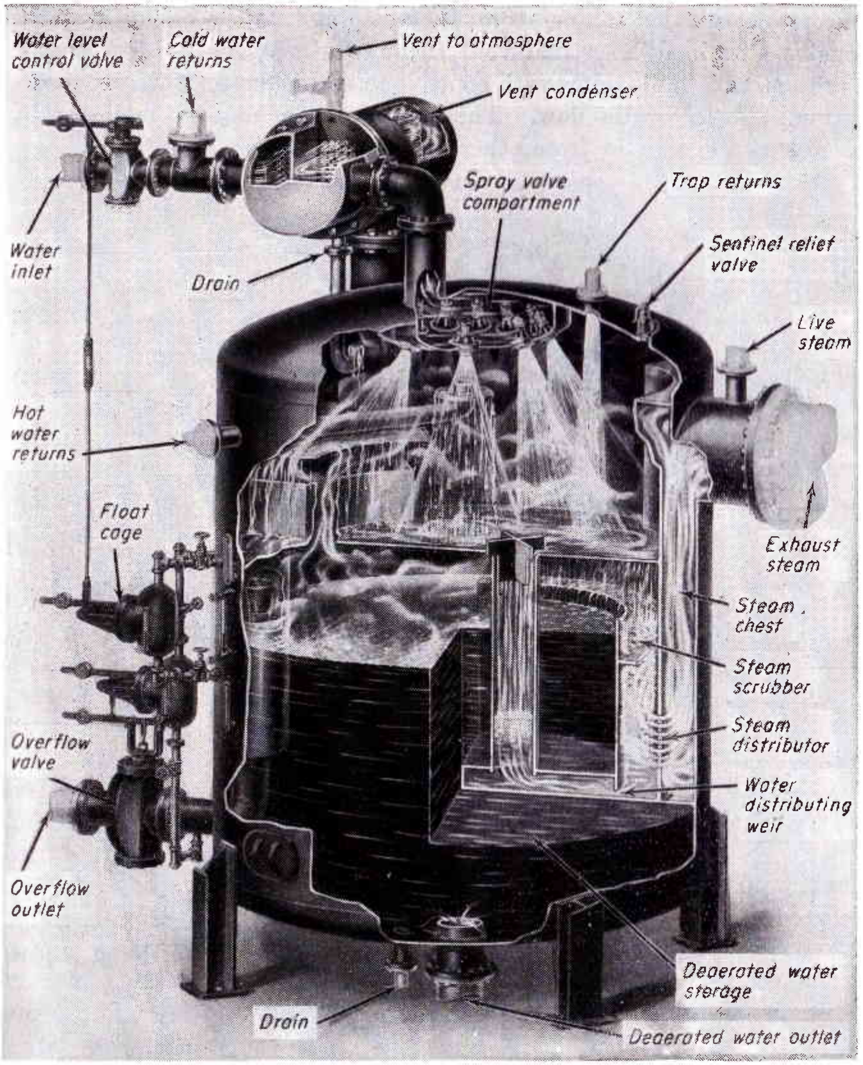


FIG. 14-1. Spray deaerator of vertical reboiling type with external vent condenser. Entering water acts as the coolant in the vent condenser and is then sprayed in a steam atmosphere and heated almost to steam temperature. Noncondensable gases are removed through a vent condenser before discharge to the atmosphere. Hot, partially deaerated water flows to the scrubber for final deaeration and then to the storage reservoir. (*Courtesy of The Permutit Company.*)

Deaeration may take place at any pressure or vacuum. Heating releases the gas from the water in minute bubbles that tend to stay entrained. These can be removed by agitating or stirring the water turbulently.

The deaerating heater and deaerator mix steam and water to heat the water to saturation temperature; they also agitate the boiling water to remove entrained gases and vent them to atmosphere. Most heaters have a storage tank below with controls to add make-up water as needed to maintain feedwater flow. The heating steam may be main-turbine bleed steam or may be from other sources.

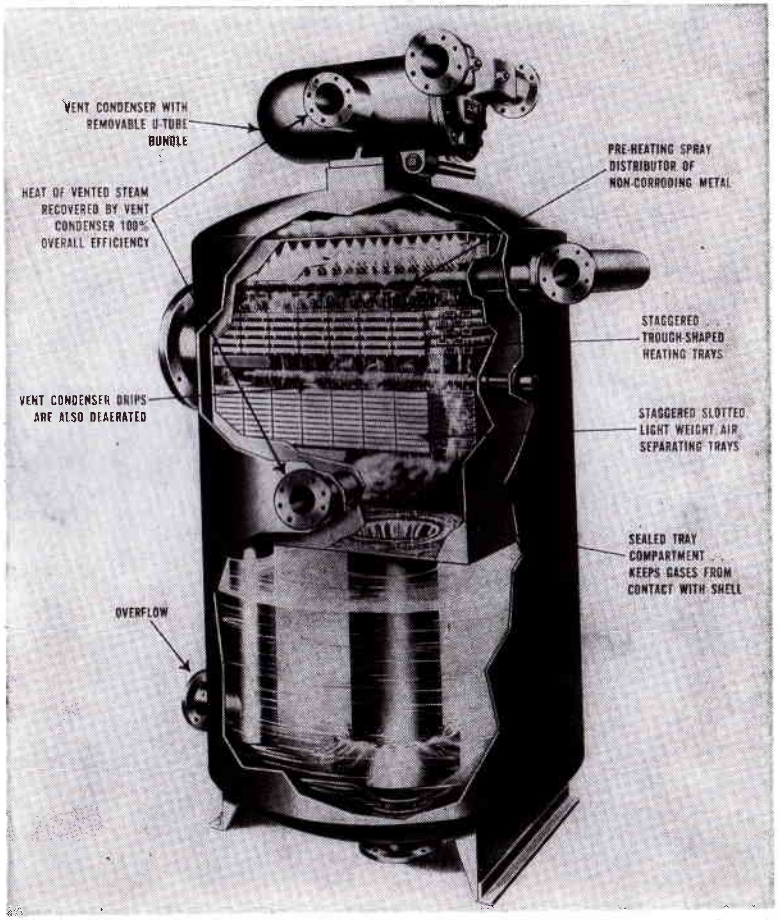

VENT CONDENSER WITH REMOVABLE U-TUBE BUNDLE
HEAT OF VENTED STEAM RECOVERED BY VENT CONDENSER 100% OVERALL EFFICIENCY
PRE-HEATING SPRAY DISTRIBUTOR OF NON-CORRODING METAL
STAGGERED TROUGH-SHAPED HEATING TRAYS
VENT CONDENSER DRIPS ARE ALSO DEAERATED
STAGGERED SLOTTED LIGHT WEIGHT AIR SEPARATING TRAYS
SEALED TRAY COMPARTMENT KEEPS GASES FROM CONTACT WITH SHELL
OVERFLOW

FIG. 14-2. Combination type of deaerator. Entering water acts as the coolant in the vent condenser and then enters the spray distributor and is sprayed upward. Heated almost to steam temperature, with most of the noncondensable gases removed, the water falls over the tray rack. Cascading through heating and air-separating trays, the water reaches steam temperature with further gas removal. It then passes into the storage reservoir. (*Courtesy of Cochrane Corporation.*)

Figure 14-1 shows a *spray type* of deaerator in which the water sprays through an atmosphere of incoming steam. A mixture of noncondensable gases and steam flows out to a vent condenser. Here the incoming cold feedwater condenses most of the steam, which flows back into the heater, and the saturated gases vent to atmosphere. Figure 14-2 shows a *combination*, or spray-and-tray, type of deaerator. The entering steam passes down at the top to meet sprayed inlet water. Both flow down over a series of two sets of trays. The upper set completes the heating process, and the lower set completes the gas separation. Released gas with some steam passes out to the vent condenser.

In a simple *tray type* of unit the feedwater after passing through the vent condenser enters the heater, falls on trays, and works its way down from tray to tray in a zigzag course, being heated and deaerated all the way.

In entering a deaerating heater steam meets the hottest water first to eliminate the last part of the gas. It carries the gas as it flows through the heater and picks up additional gas. Most of the steam condenses as it heats the water. The remaining steam, with a high gas content, reaches the top of the heater and on meeting incoming cooler water almost completely condenses. The residual steam, rich in released gases, leaves the heater for the vent condenser.

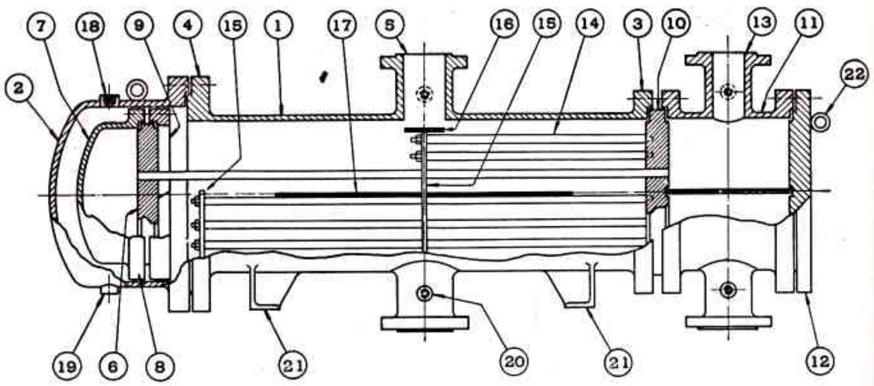
The *vent condenser* uses incoming cooler feedwater as a coolant. This condenses most of the residual steam, which flows back into the heater. Vent condensers may be direct-contact or shell-and-tube types. Positive-pressure heaters vent the gas by blowing it to atmosphere. Heaters under vacuum vent to the main condenser or use a vacuum pump.

Stainless-steel or Monel deaerator internals resist corrosion. A control valve automatically admits make-up when the heater water level falls too low or passes heated water to a storage tank if the level rises too high.

14-3. Closed Feedwater Heaters. These shell-and-tube units ordinarily use bleed steam for heating. The steam condenses on tubes carrying the feedwater. Low-pressure heaters may use steam under vacuum and high-pressure heaters at pressures above 600 psig. Steam and water need not be at the same pressure, and one pump may push water through several heaters in series. Water pressure may be as high as 4000 psig. Heater performance may be based on the first-law equation,

$$W_s(h_1 - h_2) = W_f(h_3 - h_4) \qquad (14\text{-}1)$$

where W_s = steam flow to heater, lb per hr
W_f = feedwater flow through heater, lb per hr
h_1 = inlet-steam enthalpy, Btu per lb
h_2 = leaving-condensate enthalpy, Btu per lb
h_3 = inlet-feedwater enthalpy, Btu per lb
h_4 = leaving-feedwater enthalpy, Btu per lb



1. Shell
2. Shell cover
3. Shell channel end flange
4. Shell cover end flange
5. Shell nozzle
6. Floating tubesheet
7. Floating head
8. Floating head flange
9. Floating head backing device
10. Stationary tubesheet
11. Channel
12. Channel cover
13. Channel nozzle
14. Tie rods and spacers
15. Transverse baffles or support plates
16. Impingement baffle
17. Longitudinal baffle
18. Vent connection
19. Drain connection
20. Test connection
21. Support saddles
22. Lifting ring

FIG. 14-3. Nomenclature of typical heat-exchanger parts. (*Courtesy of Tubular Exchanger Manufacturers Association, Inc.*)

If the heater receives high-pressure condensate drained from another heater, this equation must be expanded to include such items.

The heavy water tubes used in heaters make the latter contribute considerably to the total cost. Heaters use as many as six passes to place them in series. Tube arrangements include: (1) the nonremovable straight-tube bundle, (2) the hairpin bent U-tube bundle, (3) the outside packed floating head, and (4) the full internal floating head. Figure 14-3 shows a two-pass heat exchanger with an internal floating head.

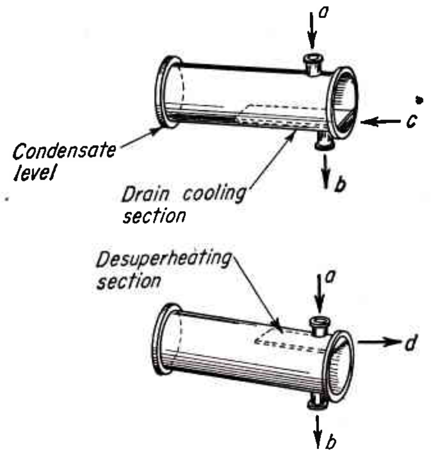


FIG. 14-4. Drain cooling and desuperheating action in a feedwater heater: *a*, incoming steam (superheated in high-pressure heater); *b*, leaving condensate; *c*, incoming low-temperature feedwater; *d*, leaving high-temperature feedwater.

Heaters also vary in shell design, channel layout, and rear-end enclosures. They may be placed vertically or horizontally. Shells have an expansion joint when tubes are welded into the tube sheets. Many heaters using superheated steam have *drain-cooling* and *desuperheating* sections. Figure 14-4 shows that by having superheated steam heat the hottest water and condensate heat the lowest-temperature water the heat-transfer irreversibility is reduced. Separate drain coolers may be used instead of integral sections. Figure 14-5 shows a modern high-pressure feedwater heater with a U-tube bundle and both desuperheating and drain-cooling sections.

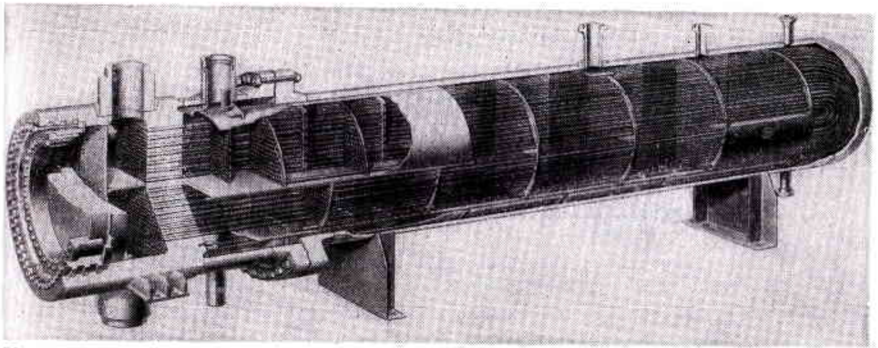


FIG. 14-5. Modern high-pressure feedwater heater with integral desuperheating and drain-cooling section. (*Courtesy of Alco Products, Incorporated.*)

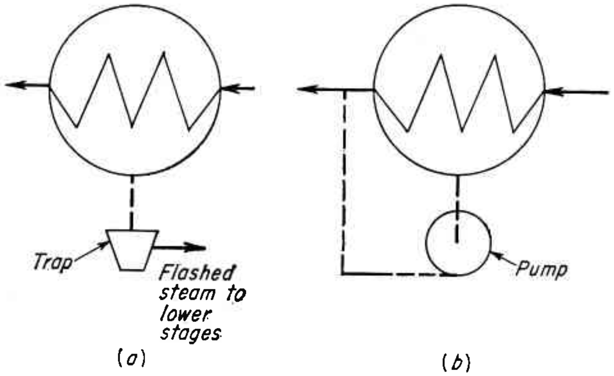


FIG. 14-6. Methods of condensate removal. (*a*) Flashing through a trap to a lower pressure; (*b*) pumping condensate into main feedwater line.

For good performance heaters must be *drained* and *vented*. Shell vents discharge gases to the next low-pressure heater or the main condenser under control of an automatic valve, for heaters working under vacuum. Level controllers maintain the heater condensate at the proper height to prevent live steam discharging through the drain or avoid the flooding of the tubes.

Heater condensate may be trapped to a lower-pressure heater, where part of it flashes into steam, or alternatively it may be pumped into the feedwater line leaving the heater (Fig. 14-6). Relief valves protect the heater shell against overpressure, as well as the tubes.

14-4. Heater Design. The Standards of the Bleeder Heater Manufacturers Association, Inc., outline the basic method of designing heaters. Figure 14-7 shows some of the recommended data to be used as follows:

$$t_{at} = t_s - 0.8t_{LMTD} \tag{14-2}$$

where t_{at} = average film temperature, °F
t_s = saturation temperature of steam, °F
t_{LMTD} = logarithmic mean temperature difference, °F

Heat-transfer rates above those for 250 F film temperature should not be used. These rates apply to steam from turbine exhaust or bleed, from evaporator outlet or boiler, and when the feedwater is like that used in a closed cycle. The standards specify that a closer approach than 2 F to saturated steam temperature should not be used in a simple condensing heater or in the condensing zone. Heat-transfer rates must be adjusted by multipliers for various conditions. If either steam or water differs in quality from that specified, the rates are multiplied by 0.85; if both differ, by 0.72.

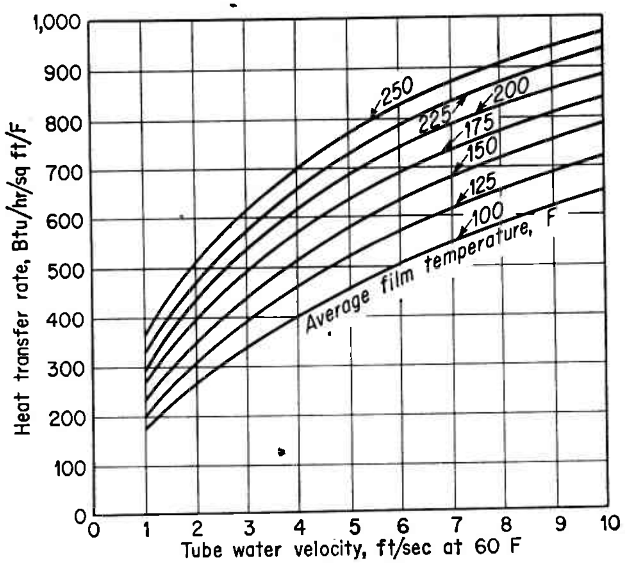


FIG. 14-7. Bleeder-heater heat-transfer rates. NOTE: Do not exceed the 250-F rate. (*Courtesy of Bleeder Heater Manufacturers Association, Inc.*)

Size and metal type need adjustments as shown in Table 14-1. Draincooling zones should be designed for a terminal difference of not less than 10 F between the shell drain and tube inlet.

TABLE 14-1. MULTIPLIERS FOR BASE HEAT-TRANSFER RATES
(For tube OD ⅝ to 1 in.)

Bwg	Tube material					
	Ars-Cu	Admiralty	90/10 Cu-Ni	80/20 Cu-Ni	70/30 Cu-Ni	Monel
18	1.00	1.00	0.97	0.95	0.92	0.89
17	1.00	1.00	0.94	0.91	0.87	0.85
16	1.00	1.00	0.91	0.88	0.84	0.82
15	1.00	0.99	0.89	0.86	0.82	0.79
14	1.00	0.96	0.85	0.82	0.77	0.75
13	0.98	0.93	0.81	0.78	0.73	0.70
12	0.95	0.90	0.77	0.73	0.68	0.65
11	0.92	0.87	0.74	0.70	0.65	0.62
10	0.89	0.83	0.69	0.66	0.60	0.58
9	0.85	0.80	0.65	0.62	0.56	0.54

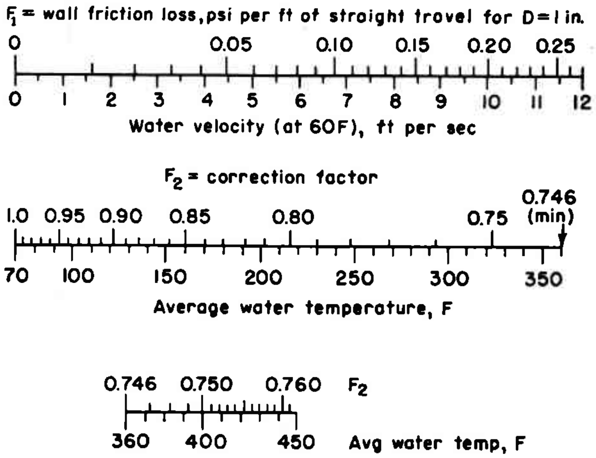


FIG. 14-8. Heater-pressure-drop correction factors. (*Adapted from Bleeder Heater Manufacturers Association, Inc.*)

The pressure drop of feedwater flowing through seamless drawn tubing in the heaters is figured as

$$\Delta p = \frac{F_1F_2(L + 5.5D)N}{D^{1.24}}$$

where Δp = pressure drop, psi
F_1 = correction factor from scale, Fig. 14-8
F_2 = correction factor from scale, Fig. 14-8
L = total lin ft of tubing divided by number of tube holes in one tube sheet

D = tube ID, in., from Table 14-2
N = number of passes

In figuring F_2 the average water temperature is equal to $t_s - t_{LMTD}$.

TABLE 14-2. FEEDWATER-HEATER TUBE CONSTANTS

Tube Bwg	⅝-in. OD		¾-in. OD		⅞-in. OD	
	D	$D^{1.24}$	D	$D^{1.24}$	D	$D^{1.24}$
18	0.527	0.452	0.652	0.589	0.777	0.731
17	0.509	0.433	0.634	0.567	0.759	0.710
16	0.495	0.418	0.620	0.554	0.745	0.695
15	0.481	0.404	0.606	0.538	0.731	0.678
14	0.458	0.380	0.584	0.514	0.709	0.652
13	.....	.....	0.560	0.488	0.685	0.625
12	.....	.....	0.532	0.457	0.657	0.594
11	.....	.....	.....	.....	0.635	0.570
10	.....	.....	.....	.....	0.607	0.538

The pressure drop through heaters is important in determining the condensate- and feed-pump pressures needed. The optimum heater arrangement has the temperature rise per heater equal to the boiler saturation temperature minus the condensate temperature divided by 1 plus the number of heaters.

Feedwater heater performance is given by: pressure drop in tubes, terminal temperature difference, and hotwell depression. Water velocities may range to above 50 ft per sec. Check valves in the bleed-steam line protect the turbine against burst water tubes. The terminal temperature difference equals the condensing temperature minus the leaving feedwater temperature. It may range from 5 to 20 F. Hotwell depression is the difference between steam saturation temperature and condensate temperature in the heater hotwell.

14-5. Miscellaneous Heaters. Water may also be heated by "waste heat" in a variety of heater arrangements. Some of these are: (1) the blowoff heat exchanger, (2) the turbine lube oil cooler, (3) generator air and hydrogen coolers, (4) the steam-jet air-ejector condenser, (5) the deaerating vent condenser, (6) the evaporator condenser, and (7) the drain cooler. These shell-and-tube heaters usually are in series with the main feedwater heaters. Heater placement depends on available temperature differences. For example, the blowoff heat exchanger using high-temperature boiler water is before or after the high-pressure heater, while the steam-jet air-ejector condenser often is the first heat exchanger in the feedwater system.

14-6. Evaporators. Evaporators produce distilled water for make-up in power plants. They prove economical where the make-up is not more than about 5 per cent of the maximum water flow.

Figure 14-9 shows a shell-and-tube evaporator that holds raw water in the shell submerging the steam-carrying tubes. The steam heat boils off the raw water to vapor, which leaves through the moisture separator and enters a separate feedwater-cooled condenser or a deaerator. This is a single-effect evaporator; they may be placed in series, the vapor from one evaporator acting as heating steam in the next one.

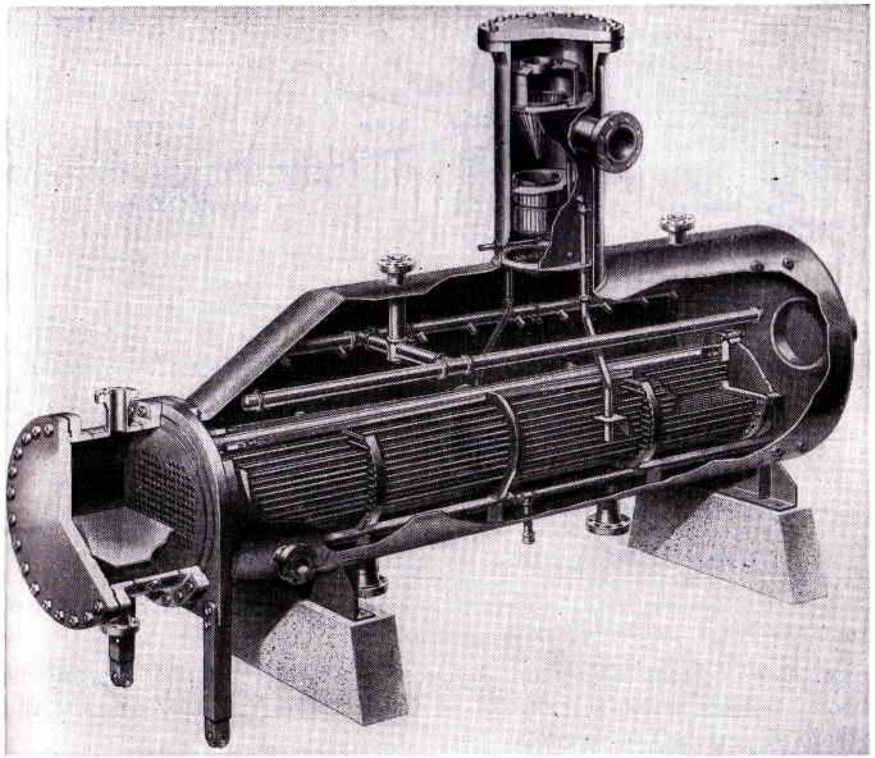


FIG. 14-9. Typical horizontal evaporator. To aid scale removal, the tubes in this design are initially bowed and in operation bend farther and so crack off scale deposits. Note the blowdown connection to keep salt concentration low and the moisture separator to prevent carry-over.

Evaporator-shell pressures vary from a vacuum to about 200 psig. The evaporator may be vertical or horizontal. Evaporators should have ample vapor-releasing surface to minimize carry-over, which carries a high salt content. Mechanical separators remove entrained drops.

Continuous evaporation of the raw water leaves behind an increasing concentration of salts in the shell. These must be blown down continuously or at intervals. Scale forming on the tubes retards heat transfer. Accumulated scale may be removed by draining the shell and spraying the tube bundle with cold water while the steam keeps the tubes hot. Thermal shock cracks off the scale, and spray washes it for removal through a bottom drain.

In Fig. 14-9 the tension bars keep the floating head from free movement and force the tubes to bow when they heat. Filling the shell quickly with cold water straightens the tubes and cracks off the scale.

14-7. Plant Layout. Figure 14-10 shows a typical utility cycle layout with four feedwater heaters, an evaporator, and miscellaneous heat exchangers. Industrial plant layouts vary widely depending on the product of the manufacturing plant and its need for electric power and steam at various pressures.

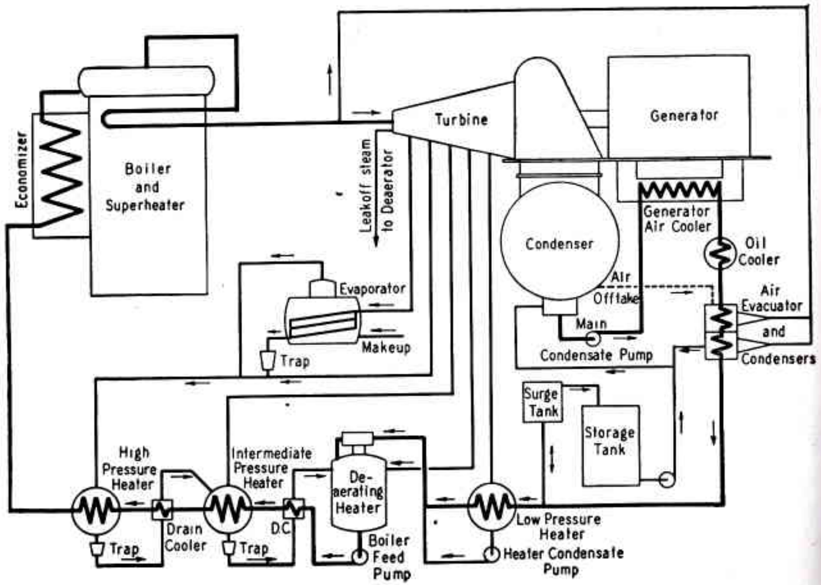


FIG. 14-10. Schematic layout of a central station to illustrate general relationship of equipment use in feed heating cycle.

PROBLEMS

14-1. Exhaust steam from an engine is used for process steam and in an open feedwater heater. Steam at exhaust has 30 per cent moisture and is at 18 psia. Feedwater rises 130 F in temperature from an inlet temperature of 60 F. Calculate the percentage of exhaust steam going to the process lines.

14-2. A closed feedwater heater receives 617,590 lb of water per hour, which increases 55.1 Btu per lb in enthalpy in passing through. The following heating flows enter the heater:

TABLE P-1

Flow, lb per hr	*h, Btu per lb*
Higher-pressure heater condensate, 37,280	350.7
Turbine gland leakage, 9,180	1366.0
Feedwater pump leakage, 63,400	355.2

Calculate the bleed steam required if bled steam at the heater is at 90.9 psia and 449 F with the condensate leaving saturated at the heater pressure.

14-3. Specify the trap capacity needed in pounds per hour for draining a closed feedwater heater if 40,000 lb of water per hour enters at 280 F and leaves at 325 F. Steam bled from the turbine is at 330 F and has 3 per cent moisture. The condensate is subcooled 10 F, and the radiation loss amounts to 5 per cent of the heat given to heat the feedwater.

14-4. Calculate the make-up water produced per hour in a double-effect evaporator operating under the following conditions:

TABLE P-2

	First effect	Second effect
Inlet water	58 F	58 F
Evaporator pressure	14.0 psia	4.0 psia
Condensate subcooled	8 F	6 F

The first effect receives 850 lb per hr of heating steam at 35 psia and a quality of 98.0 per cent.

14-5. A deaerating feedwater heater has the following flows at one point of operation:

Leaving feedwater, 800,000 lb per hr at 50 psia
Drain inflow, 72,000 lb per hr averaging 380 Btu per lb enthalpy
Entering feedwater, 175 F
Turbine bleed steam, 50 psia, 320 F

Calculate the amount of (*a*) bled steam from turbine; (*b*) entering feedwater.

14-6. A feedwater heater is to be designed for the following conditions:

Bled steam, 33,000 lb per hr, 600 psia, 530 F
Entering feedwater, 455 F
Terminal temperature difference, 4 F
Condensate subcooling, 10 F
Tube data, ⅞-in.-OD Monel metal, 14 Bwg, water velocity 540 ft per min

Calculate the tube length for a two-pass heater.

14-7. Sketch a flow diagram for a plant with the following equipment and flow paths:

Turbine:
  No. 1 bleed to high-pressure closed heater
  No. 2 bleed to intermediate-pressure closed heater
  No. 3 bleed to deaerator
  No. 4 bleed to low-pressure closed heater
Heater No. 1 and 2 condensate flashed to next heater; heater No. 4 condensate pumped
Evaporator, steam from No. 2 turbine bleed, discharge to deaerator, drain to deaerator
Miscellaneous heaters, hydrogen cooler, ejector condenser
Major pumps, main condensate, boiler feed takes suction from deaerator

14-8. Sketch a flow diagram for a double reheat plant with the following equipment and flow paths:

Turbine, superpressure, very-high-pressure, high-pressure, double-flow intermediate-pressure, and double-flow low-pressure sections
Reheater, one between very-high-pressure exhaust and high-pressure inlet, second between high-pressure exhaust and intermediate-pressure inlet
Pumps. Motor-driven condensate pump discharges through drain cooler, gland condenser, 1st heater, drain cooler, stack economizer, 3d heater to 4th deaerator; motor-driven low-pressure boiler feed pump discharges through 5, 6, 7, 8, 9 closed heaters in series to intermediate-pressure motor-driven boiler-feed-pump inlet, which in turn discharges to a steam-turbine-driven high-pressure boiler feed pump
Heaters:
  1st heater receives steam from low-pressure bleed point, condensate to drain cooler then flashed to condenser; 1st heater also has a hot-water coil to supply heat to air tempering heater
  2d heater is a stack economizer
  3d heater receives steam from low-pressure bleed point
  4th heater receives steam from intermediate-pressure bleed
  5th heater receives steam from intermediate-pressure bleed
  6th heater receives steam from high-pressure exhaust before reheat and from feed-pump turbine exhaust
  7th heater receives steam from high-pressure bleed
  8th heater receives steam from high-pressure bleed
  9th heater receives steam from very-high-pressure exhaust before reheat; the boiler feed-pump turbine is also supplied from the very-high-pressure exhaust
All heater condensate flashed to next lower heater except that from No. 5, which is pumped to deaerator

14-9. A low-pressure feedwater heater is designed for condensate that enters at 81.0 F and leaves at 179.8 F. Extracted steam at the rate of 65,400 lb per hr and enthalpy of 1185 Btu per lb enters the heater, together with a flow of 77,500 lb per hr of drain at h = 173.4 Btu per lb from the higher-pressure heater. Enthalpy of the shell-side liquid is 152.5 Btu per lb when leaving. The LMTD is 32 F, and the over-all heat-transfer rate is 640 Btu per hr-sq ft-°F.

a. What is the temperature of the condensing steam?
b. At approximately what pressure is the heater operating?
c. How many pounds per hour of condensate are flowing?
d. What surface area is required?
e. If the velocity through the tubes is 5.9 ft per sec, compute the number and length of ⅝-in.-OD. No. 18 Bwg tubes required.

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CHAPTER 15

POWER-STATION PUMPS

15-1. Pump Applications. Power plants need pumps for various services with the general features shown in the following table:

Service	Head	Capacity
Boiler feed	High	Medium
Circulating water	Low	High
Condensate	Medium	Medium
Heater drains	High	Low
Service water	Medium	Medium
Treated water	Medium	Medium
Evaporator feed	High	Low
Well water	Medium	Medium
Ash sluicing	Medium	Medium
Chemical feed	Medium	Low
Fuel oil	Medium	Low
Liquid metal	Medium	High

These services need a variety of designs using different principles of operation and structural details.

15-2. Pump Features. Power-plant pumps may be classified as in the following table:

Class	Type	Class	Type
Reciprocating	Direct-acting	Rotary	Gear
	Power		Lobe
Centrifugal	Volute		Vane
	Diffuser		Screw
	Mixed flow		Cam and piston
	Axial or propeller flow		Others
	Turbine or regenerative		

Performance curves and efficiency data determine the power input needed to run the pump under given conditions. For liquid pumps the input is

$$W_i = \frac{m(\Delta FE + \Delta KE + \Delta PE)}{550} \qquad (15\text{-}1)$$

where W_i = ideal pump or water horsepower, hp
m = mass fluid pumped, lb per sec
ΔFE = flow-energy or static-pressure head change, ft-lb per lb fluid
ΔKE = kinetic-energy or velocity head change, ft-lb per lb fluid
ΔPE = potential-energy change or elevation change, ft-lb per lb fluid

Energy changes are + for increases and − for decreases.

The actual pump input power is

$$W_a = \frac{W_i}{\eta_p} \tag{15-2}$$

where W_a = actual power input, hp
η_p = pump efficiency, fraction

The difference $W_a - W_i$ appears as increased internal energy in the fluid and minor losses. Geometrically similar pumps at equal efficiencies are related as follows

$$H = K_1 N^2 D^2 \qquad Q = K_2 N D^3 \qquad P = K_3 \rho N^3 D^5 \tag{15-3}$$

where K = constant
H = total head, ft of fluid
N = speed, rpm
P = power, hp
D = characteristic dimension (usually outlet diameter)
ρ = fluid density, lb per cu ft

Performance at different conditions may be analyzed with these equations. For example, for a given pump at two different speeds, N_1 and N_2,

$$\frac{H_1}{H_2} = \frac{N_1^2}{N_2^2} \qquad \frac{Q_1}{Q_2} = \frac{N_1}{N_2} \qquad \frac{P_1}{P_2} = \frac{N_1^3}{N_2^3} \tag{15-4}$$

Centrifugal pumps have a parameter called *specific speed,*

$$N_s = \frac{NQ^{1/2}}{H^{3/4}} \tag{15-5}$$

where N_s = specific speed, rpm
N = speed, rpm
Q = flow rate, gpm
H = total head or energy, ft of fluid

Specific speed is a characteristic number that represents the rpm at which an impeller would run if reduced in size to deliver 1 gpm against a total head of 1 ft. This number is an index of pump type, using the head obtained for a given capacity at maximum efficiency. Impellers for high heads usually have low specific speed; impellers for low heads have high specific speeds. Figure 15-1 shows the general impeller shape for given specific speeds.

The liquid flowing into a pump has a definite vapor pressure at a given temperature. If the pressure of the liquid in passing through the impeller at any point drops to the vapor-pressure level, bubbles of vapor and air form at this point. As the liquid and vapor mixture flows over the impeller surface to a higher-pressure region, the bubbles collapse suddenly and the liquid pounds the impeller with high pressure or impact. These act with high frequency on an impeller, forming pits by erosion. This action, called *cavitation,* lowers efficiency, makes the pump noisy, and damages the pump.

To avoid flashing and cavitation, the liquid must have a certain *net positive suction head* (NPSH) at the pump inlet, referred to the pump center line. This is the energy in feet of liquid pumped that must exist in the system inlet over and above the vapor pressure of the liquid. The available NPSH is

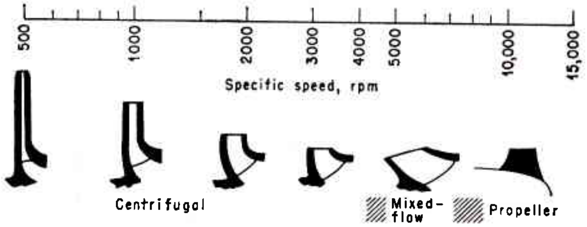


FIG. 15-1. General relation of specific speed and impeller shape for centrifugal pumps.

$$NPSH = \frac{p_1 - p_v}{\rho} + (S - h_f) \tag{15-6}$$

where $NPSH$ = available net positive suction head during operation, ft of fluid
p_1 = pressure on surface of liquid in *suction vessel,* psf
p_v = vapor pressure at liquid temperature, psf
ρ = fluid density, lb per cu ft
S = distance from pump center line to level of liquid in suction vessel, ft, plus if level above pump, minus if below
h_f = friction losses in suction piping, ft of fluid

This concept is used in a cavitation parameter called the "Thoma coefficient,"

$$\sigma = \frac{NPSH}{H} \tag{15-7}$$

The magnitude of the coefficient when cavitation just starts is the critical value. Pumps should be installed to keep the coefficient higher than the critical. These are found experimentally.

A deaerating-heater pump often runs into cavitation. If the deaerator

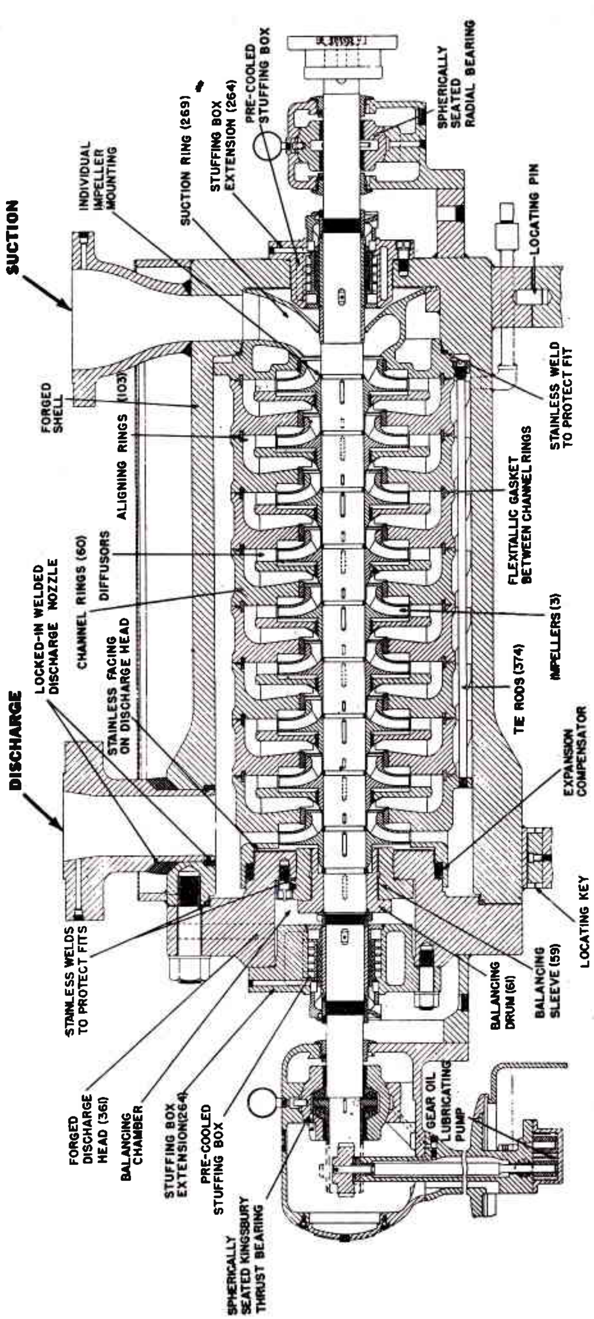

SUCTION
DISCHARGE
INDIVIDUAL IMPELLER MOUNTING
SUCTION RING (269)
STUFFING BOX EXTENSION (264)
PRE-COOLED STUFFING BOX
SPHERICALLY SEATED RADIAL BEARING
LOCATING PIN
STAINLESS WELD TO PROTECT FIT
FLEXITALLIC GASKET BETWEEN CHANNEL RINGS
IMPELLERS (3)
TIE RODS (374)
EXPANSION COMPENSATOR
LOCATING KEY
BALANCING SLEEVE (59)
BALANCING DRUM (61)
GEAR OIL LUBRICATING PUMP
SPHERICALLY SEATED KINGSBURY THRUST BEARING
PRE-COOLED STUFFING BOX
STUFFING BOX EXTENSION(264)
BALANCING CHAMBER
FORGED DISCHARGE HEAD (36)
STAINLESS WELDS TO PROTECT FITS
LOCKED-IN WELDED DISCHARGE NOZZLE
CHANNEL RINGS (60)
DIFFUSORS
STAINLESS FACING ON DISCHARGE HEAD
ALIGNING RINGS (103)
FORGED SHELL

FIG. 15-2. Typical high-pressure boiler feed pump for subcritical power plants. Note pumping element of impellers, channel rings, and shaft encased in forged steel casing. Discharge pressure seats the channel rings. Axial balance is obtained by a balancing drum. Momentary unbalance is absorbed by a Kingsbury thrust bearing. (*Courtesy of Ingersoll-Rand Company.*)

cannot be located high enough above its drain pump to develop a high enough NPSH, other methods must be used. These include injecting colder water into the pump suction or placing a condensate-cooled heat exchanger in the suction line. The latter may raise the friction loss and tend to lower the NPSH.

15-3. Boiler Feed Pumps. Small low-pressure plants sometimes use direct-acting duplex reciprocating pumps. For higher pressures the triplex pump may be used. Centrifugal pumps, however, are generally preferred today. They develop continuous flow, run at high speeds, and are compact.

These pumps usually have more than one rotor stage for raising pressure in steps. Figure 15-2 shows one type of high-pressure feedwater pump; the multistage impeller turns inside a casing, which must have seals at the shaft penetrations. The rotor must be perfectly balanced and restrained by a thrust bearing.

Axial thrust may be balanced by: (1) a balancing disk or drum, (2) double suction impellers on all stages, (3) dividing an even number of stages into two opposing groups, and (4) balancing every stage by a balancing chamber on the back of each.

The feed pump becomes a major power consumer in the plant cycle as the boiler pressure increases. At 2,400 psi the feed pump uses about $3\frac{1}{2}$ per cent of the gross plant capability, and at 5,500 psi it requires about 6.0 per cent. At high enough pressures (perhaps 80,000 psi) it may require the entire plant output. Designers estimate that 10,000 psi will be the practical limit. High-pressure plants use centrifugal feed pumps with a double-casing barrel type of design (Fig. 15-3). Feed-pump design has advanced by using high-speed rotors and impellers to reduce the physical length of the rotor between bearings. Superpressure plants must have rotor speeds as high as 9,000 rpm to develop the needed feedwater pressure.

Designs vary in the number of pumps used in a plant cycle. Some use three half-sized pumps for capacities above 100,000 kw; two run, while the third acts as stand-by. Others use two full-sized pumps, one acting as stand-by for emergencies. Again, some designers prefer primary and secondary pumps with closed heaters between, rather than one pump discharging through several heaters downstream.

The pump work also raises feedwater temperature. Neglecting bearing losses and radiation, the temperature rise can be estimated as

$$\Delta t = \frac{H}{778}\left(\frac{1}{\eta_p} - 1\right) \tag{15-8}$$

where Δt = temperature rise, °F
H = pump head, ft
η_p = pump efficiency, fraction

For pumps having a balancing device with leak-off the above relation may be used if η_p is increased by the ratio

$$R = \frac{Q + Q_1}{Q} \tag{15-9}$$

where Q = pump discharge, lb per hr
Q_1 = leak-off, lb per hr

15-4. Boiler-feed-pump Drives. Most plants use direct-connected electric drives up to 3,600 rpm in the form of constant-speed squirrel-cage induction motors. This requires throttling the pump discharge to regulate flow. Variable speed may be used instead with greater efficiency by using magnetic or hydraulic couplings to a constant-speed motor.

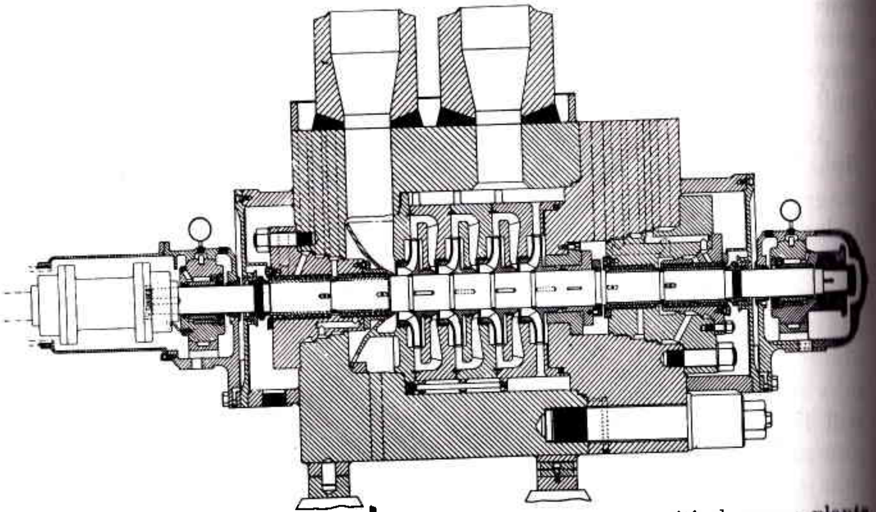


FIG. 15-3. Typical high-pressure boiler feed pump for supercritical power plants. This installation uses three half-capacity pumps in each of two circuits in which the above is the *high-pressure unit*. The pumps are designed to deliver 2×10^6 lb per hr at 570 F and 6,500 psi. (*Courtesy of Ingersoll-Rand Company.*)

Steam turbines widely used in older plants for pump drives were displaced by motors. But the rising pressures of modern plants make the high speed of turbines a favorable factor, and so they are again being used. The turbine reduces the power drain on the main generator, making more capacity available to carry load. Turbine drives cost less than electric motors at the higher powers. The turbine takes steam from an extraction stage or from a cold-reheat line and exhausts to a low-pressure heater.

15-5. Condensate and Heater Drain Pumps. Condensate pumps take their suction from the hotwell of the condenser. The condensate is at saturation temperature in the hotwell, so the pump must be below the hotwell to assure adequate NPSH. The pump discharges through air-ejector condensers, H_2 coolers, and several feedwater heaters. The water then goes to the deaerating heater or the boiler-feed-pump suction.

Figure 15-4 shows a two-stage horizontal condensate pump with the stage inlets opposed. This helps the axial balance, and the sealing glands both have a positive internal pressure that prevents air leaking into the casing. The bearings mount externally to prevent oil leaking into the condensate. These pumps may stand vertically or horizontally. In vertical mounting only the pump need be below the hotwell. The motor drive above makes maintenance easier and keeps the motor safe from possible floods in the pump pit. Small constant-speed or squirrel-cage induction motors usually drive condensate pumps. For large units a wound-rotor induction motor or a squirrel-cage induction motor with hydraulic coupling gives variable-speed drive. Some plants use steam-turbine drives.

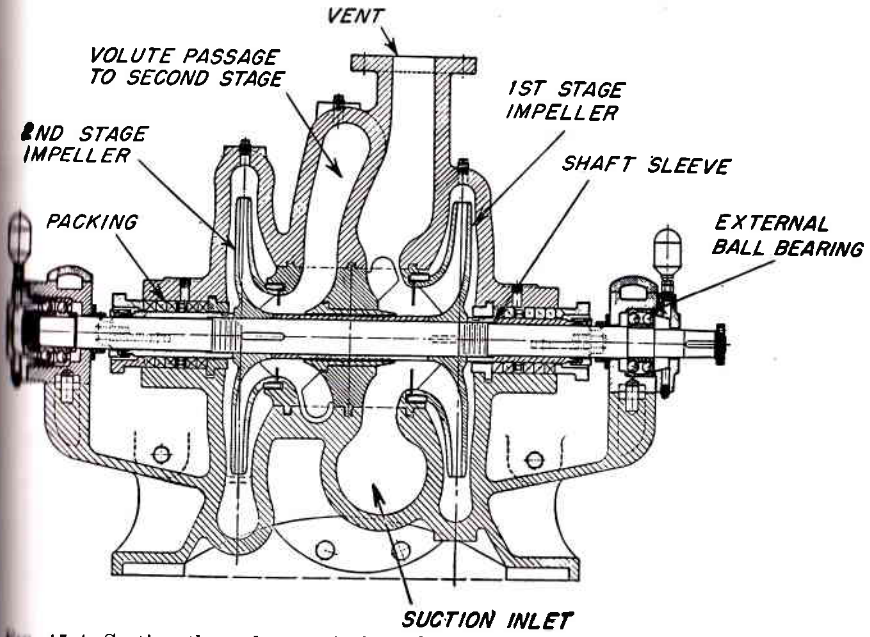


FIG. 15-4. Section through a typical condensate pump. Note that both glands are subjected to high-pressure liquid to prevent air leakage. (*Courtesy of Westinghouse Electric Corporation.*)

Heater drain pumps need only low capacity but must develop high pressures. Figure 15-5 shows a single-stage heater drain pump.

15-6. Circulating Water Pumps. These pumps run against a low head and handle large volumes of water. Condenser circulating pumps

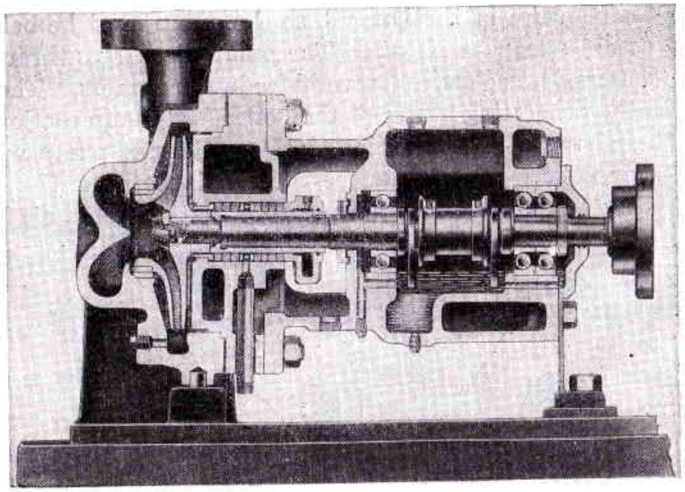


Fig. 15-5. Single-stage heater drain pump. (*Courtesy of Ingersoll-Rand Company.*)

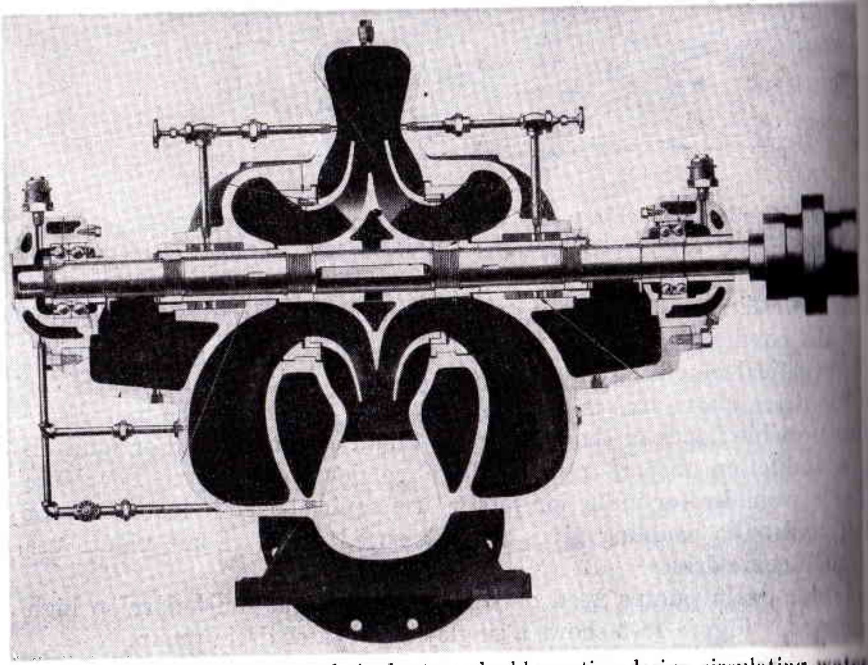


Fig. 15-6. Horizontally mounted single-stage double-suction-design circulating water pump. (*Courtesy of Worthington Pump and Machinery Corporation.*)

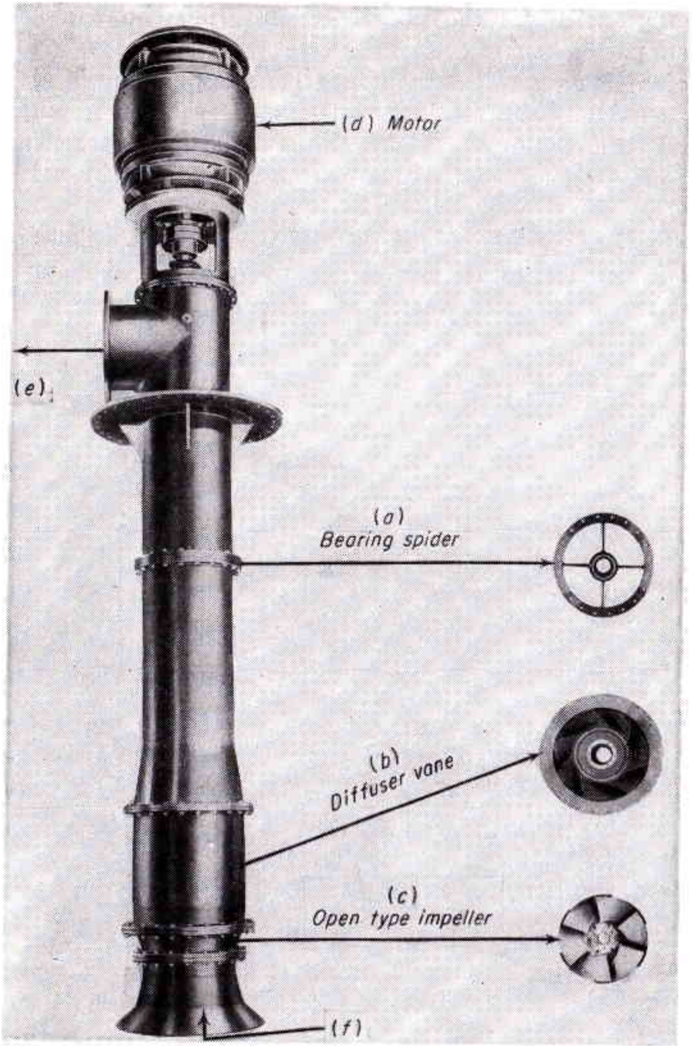


Fig. 15-7. Mixed-flow-type condenser circulating water pump: *a*, bearing spider gives intermediate bearing support; *b*, diffuser vanes straighten water flow from impeller; *c*, impeller of mixed-flow type; *d*, motor drive; *e*, discharge; *f*, inlet. (*Courtesy of Allis-Chalmers Manufacturing Company.*)

are made in three types: (1) centrifugal volute, (2) mixed flow, and (3) propeller.

The *centrifugal volute* pump can be used for all applications but does best for high suction or pumping heads. Such pumps usually have a single-stage double-suction bronze impeller mounted on a steel shaft in a horizontally split cast-iron casing. Slow-speed motors or a turbine with reduction gears drives the pump. Figure 15-6 shows a centrifugal volute pump.

The *mixed-flow* pump best handles moderate pumping heads of 20 to 50 ft and suction lifts. The lower casing of the submerged pump runner forms the suction piping and the upper casing the discharge. The driving motor mounts on top of the pump, connecting to it by a long shaft (Fig. 15-7).

The *propeller* type works best with large to moderate flows at pumping heads lower than for the mixed flow, up to 20 ft. Some pumps have variable-pitch blades that can be adjusted to suit changes in the water circuit as needed.

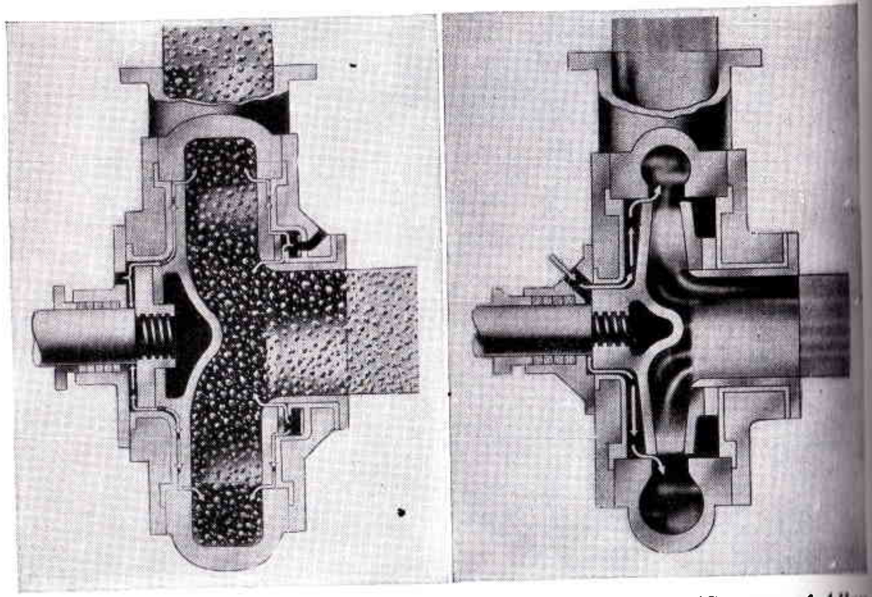


FIG. 15-8. Pump for handling abrasive solids. Note seal design. (*Courtesy of Allen-Sherman-Hoff Co.*)

15-7. Miscellaneous Pumps. A power plant needs a variety of pumps for certain special services.

Ash-handling Pumps. These handle air- or water-borne ash particles in disposal systems. They may be centrifugal pumps with special seals and rubber lining in the casing (Fig. 15-8).

Chemical Feed Pumps. These inject dissolved chemicals into feed-water systems and boiler drums. They must have very flexible capacity control to regulate discharge rates exactly. Figure 15-9 shows a pump with reciprocating pumping elements.

Service Water Pumps. These supply water to hydrogen coolers, bearing cooling, air-compressor cooling, oil coolers, and many others. These

pumps draw water from the intake tunnel or circulating-pump discharge. Most plants have several service pumps fitted with strainers to remove debris from the suction water. Most pumps are the single-stage split-case type.

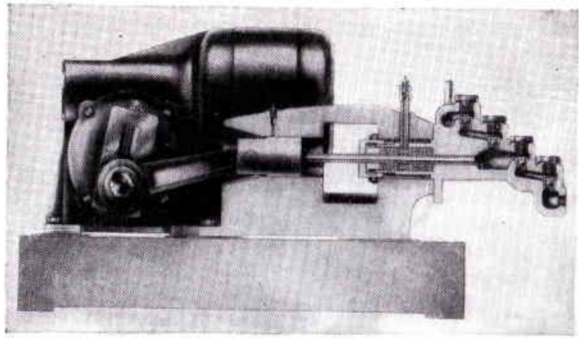


FIG. 15-9. Reciprocating type of chemical feed pump. Note ball-type construction of suction and discharge valves. (*Courtesy of Milton Roy Company.*)

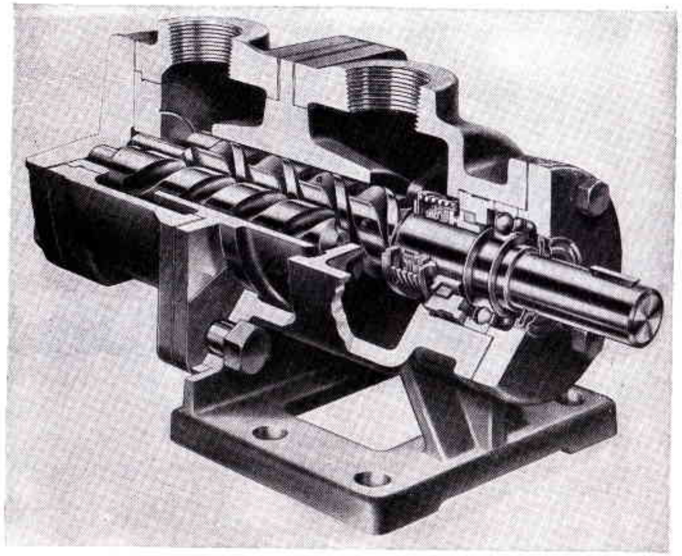


FIG. 15-10. Typical screw-type fuel pump. Pumps of this type are also used in other applications, as lubrication, circulating, etc. (*Courtesy of DeLaval Steam Turbine Co.*)

Evaporator Feed Pump. This supplies raw water to the evaporator shell. Make-up requirements up to 5 per cent of station maximum flow set the pump capacity needed. Constant-speed single-stage pumps are favored.

Fuel-oil Pumps. These transfer oil from the storage tank to day tanks and pressurize oil on its way to the burners. Many modern plants use the screw or gear type of pump (Fig. 15-10). Some plants use direct-acting reciprocating steam pumps.

Liquid-metal Pumps. These are used in some nuclear power plants to handle bismuth, sodium, sodium-potassium, and other metals. Some

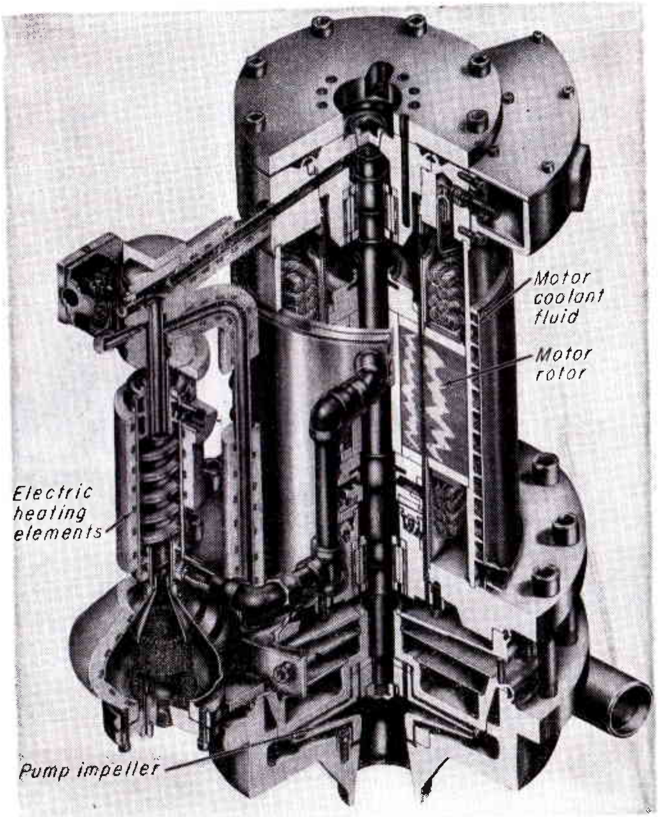
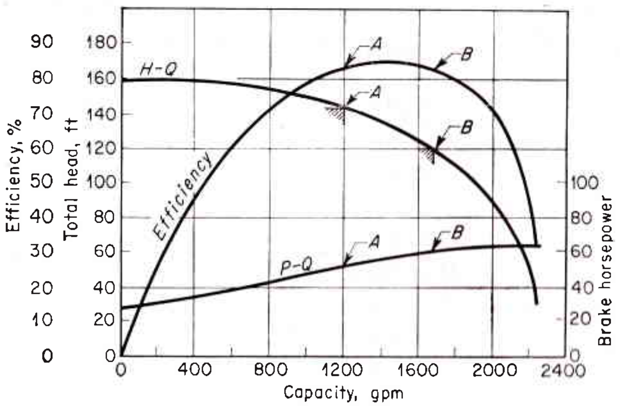
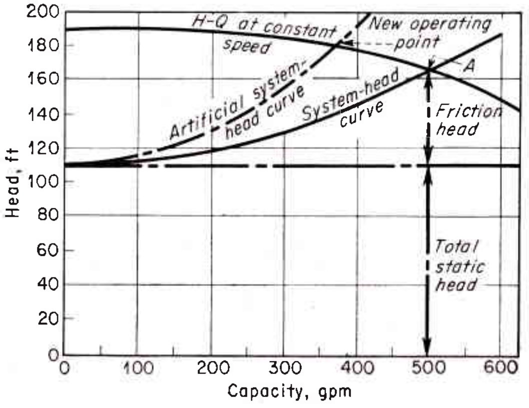


Fig. 15-11. Hermetically sealed, or "canned," type of centrifugal liquid-metal pump. (*Courtesy of Westinghouse Electric Corporation.*)

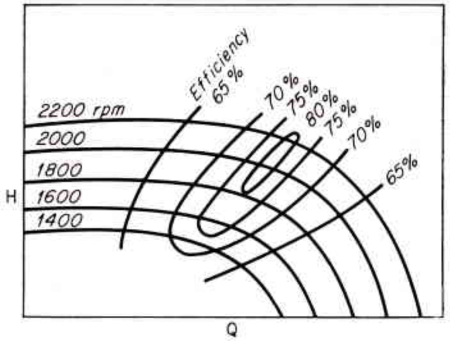
resemble conventional centrifugal pumps, and others work on electromagnetic principles without any mechanical moving parts. Figure 15-11 shows a hermetically sealed centrifugal pump used for this service. In the electromagnetic pump a flow duct is surrounded by a field winding. When an alternating current energizes the winding, it induces an alternating current in the liquid metal in the flow duct. The induced current has its own magnetic field, which reacts with the field of the winding to force the liquid-metal particles to move against a head.



(a) Characteristic curves



(b) System head curve



(c) Variable speed curves

Fig. 15-12. Representative centrifugal-pump and system curves. These curves are intended to show only general trends. The manufacturer must be consulted for exact data for any pump under consideration.

Deep-well Pump. This has a narrow diameter designed to fit in a deep well bottom and pump water to the surface. The pump has a vertical impeller of several stages and may work at depths of as much as 2,000 ft.

Treated-water Pumps. These handle the water undergoing purification in a water-treatment system of a power plant. They are usually one- or two-stage pumps driven by constant-speed motors.

15-8. Pump Performance. Positive-displacement pumps usually have an approximately constant capacity at any given speed, with the capacity decreasing slightly as the discharge pressure increases. A centrifugal pump may have a capacity range from zero flow to maximum at any one speed (Fig. 15-12). The *characteristic curves* show the effect of capacity on head, power, and efficiency at constant speed. A pump may run at any two points, as *A* or *B*. At any such point it must have three consistent values of head, power, and efficiency as shown. The pump can run only on its characteristic.

The system to which a pump discharges has a *system head curve* that shows the amount of head needed to move a given flow through it. The system head curve includes the friction head curve, static head, and any pressure differences in the system. Friction head curves show friction losses through pipes, valves, and fittings at various flow rates. Static head is the difference in elevation between suction and discharge. If a valve in the discharge line can be varied, an artificial system head curve for each valve opening can be plotted. This shows how flow may be controlled by a discharge valve; the intersection of the system head and pump head curves gives the operating condition for the system. *Variable-speed curves* give the pump behavior as impeller speed changes. These curves also show efficiencies.

PROBLEMS

15-1. Data from a turbine and condenser are as follows:

Turbine:
  Three-stage 75,000-kw bleeder condensing unit
  Inlet-steam pressure, 330 psia; temperature, 720 F
  Steam flow at full load, 800,000 lb per hr
  Mass and enthalpy of bled steam as in Table P-1

Table P-1

Bleed point	1	2	3
Lb per hr	47,450	78,980	20,100
h, Btu per lb	1287	1203	1087

  Radiation and friction loss, 0.3 per cent of turbine output
Condenser:
  Pressure, 2.5 in. Hg abs
  Condensate subcooled 10 F
Circulating pumps:
  Two centrifugal pumps
  Pump efficiency, 90 per cent; pump head, 39.3 ft
  Condensing-water temperature rise, 9 F
Hotwell pump:
  One centrifugal pump
  Pump efficiency, 85 per cent; pump head, 86 ft

Draw a line diagram of the above equipment, and calculate:

a. Horsepower input and the gallons per minute of each circulating water pump
b. Horsepower input to hotwell pump
c. Total pump power as a percentage of turbine output

15-2. Calculate the efficiency of a centrifugal pump under the following conditions:

Motor input, 21.1 kw
Suction pressure, 5.0 in. Hg vacuum in a 10-in.-diameter pipe
Discharge pressure, 25.0 psig in a 6-in.-diameter pipe with gauge 7 ft above suction gauge
Water flow, 920 gpm at 70 F

15-3. Calculate the motor capacity needed for a boiler feed pump taking water from an open heater and discharging to a boiler under the following conditions:

Boiler pressure, 450 psia
Heater pressure, 15 psia
Discharge-line pressure drop, 80 psi
Pump efficiency, 82 per cent
Water flow, 325,000 lb per hr

15-4. A centrifugal pump gave the following performance:

Total head, 82 ft
Water flow, 685 gpm at 180 F
Speed, 1,475 rpm
Motor input, 13.2 kw

a. Calculate the pump efficiency.
b. Estimate the discharge head if the pump speed were reduced to 800 rpm.

15-5. A boiler feed pump receives 705 gpm of water at 374.2 F and an enthalpy of 348.7 Btu per lb. It operates against a head of 3,120 ft with an efficiency of 61.8 per cent.

a. Calculate the enthalpy leaving the pump in Btu per pound.
b. Estimate the leaving-water temperature, assuming the temperature rise as due to the inefficiency of the input energy.
c. Calculate the discharge pressure in pounds per square inch absolute for a suction pressure of 600 psia.
d. Calculate the power output of the driving motor in kilowatts.

15-6. A deaerating heater operating at 50 psia is piped to the suction of a boiler feed pump vertically below. A Thoma coefficient of 0.3 is to be designed for, to avoid cavitation. At the design flow a friction loss of 30 ft per 1,000 ft of pipe and a velocity head of 1.8 ft occurs. Estimate the minimum vertical run of pump suction piping required for a discharge pressure of 600 psia. Analyze your answer, and suggest proposals for the installation.

15-7. A pump is to discharge 200,000 gpm at a head of 14 ft when running at 360 rpm. What type of pump is required?

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CHAPTER 16

POWER-STATION PIPING ELEMENTS AND HEAT BALANCE

16-1. Piping Systems. Power plants use many fluids in their operation—steam, water, oil, air, gas, etc.—that must be completely controlled and coordinated. This leads to a variety of integrated piping systems that must be run between and around the main and auxiliary equipment. Some of these piping systems are:

1. Steam piping—main, auxiliary, reheat, bleed, exhaust, heating steam, process steam
2. Water piping—condensate, feedwater, raw water
3. Blowoff piping—boiler, evaporator, feed treatment
4. Condenser cooling water
5. Miscellaneous piping—service water, water treatment, drains, fuel, lubricating oil, compressed air, vent, vacuum, soot blowing, fire protection, generator hydrogen cooling, chemical feed, etc.

Power-plant piping would be simple if the plant always ran at a continuous constant load. But the need for placing equipment in service, running it at varying outputs, and taking it out of service for inspection and repairs makes the piping layout very complex and costly. Piping in a power plant may represent as much as $12 to $20 per kilowatt.

16-2. Pipe Size and Pressure Loss. As fluids flow through pipes and ducts, they lose pressure from frictional effects and interchange heat with the pipe and surroundings. Section 4-17 shows how to figure pressure losses for a given pipe size and flow pattern. Equation (4-51) shows that pressure losses decrease as piping diameter increases. But larger pipe is more expensive and may increase heat losses. The first cost of piping must be balanced against the value of reducing pressure losses—which sometimes means the cost of pumping, or the loss of available energy. Pipe diameter affects the fluid speed for a given mass flow rate (Table 16-1).

16-3. Pipe Strength. Power-station piping uses carbon-steel material for temperatures up to 850 F. Higher temperatures require alloy steels that include chromium, vanadium, molybdenum, and nickel. Chromium improves corrosion and oxidation resistance; nickel adds toughness;

molybdenum improves creep strength. Carbon-molybdenum steels are used for ranges of 750 to 1000 F and chromium steels for higher temperatures.

A group of alloy steels, called "stainless steels," have a high chromium content of the order of 16 per cent. Ordinary carbon steels and the stainless steels with more than 16 per cent chromium are varieties of ferritic steel. Nonmagnetic stainless steels are austenitic materials.

TABLE 16-1. RECOMMENDED PIPE VELOCITIES

Pipeline	*Velocity, ft per min*
Boiler and turbine leads	6,000–12,000
Steam headers	6,000– 8,000
Branch steam lines	6,000–15,000
Feedwater lines	240– 840
Exhaust and low-pressure steam lines	6,000–15,000
Pump suction lines	120– 300
Bleed lines	4,000– 6,000

Various codes and standards specify minimum pipe dimensions and materials to meet certain safety requirements. Those most used for station piping are the ASME Boiler and Pressure Vessel Code and the ASA B31.1 Code for Pressure Piping.

The design of pipe sizes must anticipate many stress-inducing factors. The principal ones include: (1) internal (or external) pressures; (2) thermal-expansion restraints; (3) extraneous movement of supports and foundations; and (4) dead weight of pipe, fittings, insulation, etc. Pipe wall thickness to meet code requirements may be calculated from

$$t_m = \frac{DP}{2(S + YP)} + C \tag{16-1}$$

where t_m = minimum wall thickness, in.
D = outside diameter, in.
P = design pressure, psi
S = allowable stress, psi
Y = coefficient as in Table 16-2
C = allowance for threading, mechanical strength, and corrosion, in.

TABLE 16-2. VALUES FOR COEFFICIENT Y

	Temperature, °F					
	900 and below	950	1000	1050	1100	1150 and above
Ferritic steel	0.4	0.5	0.7	0.7	0.7	0.7
Austenitic steel	0.4	0.4	0.4	0.4	0.5	0.7

The value of C for various conditions is as shown in the following table:

Pipe	*C, in.*
Plain end steel or wrought-iron pipe 1 in. and smaller	0.05
Plain end steel or wrought-iron pipe larger than 1 in.	0.065
Grooved steel, wrought iron, or nonferrous pipe	Depth of groove

Since pipe mills do not fabricate to precise sizes, a tolerance above or below the pipe size is required. Allowance must be made for this in specifying wall thickness from Eq. (16-1), on the order of a 12½ per cent increase.

The codes should be consulted for allowable stress values. They depend on temperature and vary for different types of steel. Pipe thickness [Eq. (16-1)] should be increased to allow for mill tolerance; usually the next largest nominal pipe diameter is chosen if the calculated thickness does not correspond to a standard size. Table 16-3 is adapted from the codes and lists allowable stress values for several pipe materials. Table 16-4 gives the dimensions for welded and seamless steel pipe. Pipe is fabricated according to a system of *nominal* (approximate) sizes, corresponding to outside diameters. Pipe thickness for all diameters is designated by *schedule numbers.* Stress and schedule numbers are related by

TABLE 16-3. ALLOWABLE S VALUES FOR PIPE IN POWER PIPING SYSTEMS[a]

Material	Specifications and grade	Maximum S for given temperatures, psi				
		−20–650 F	800 F	900 F	1000 F	1100 F
Electric-resistance welded steel	A53, A135; A	10,200	7,650	4,250		
	A53, A135; B	12,750	9,200	4,250		
Electric-fusion welded steel	A155, A285; A	10,100	7,500			
	A155, A285; B	11,250	8,450			
Seamless carbon steel	A53, A106; A	12,000	9,000	5,000	1,500	
	A53, A106; B	15,000	10,800	5,000	1,500	
Seamless alloy steel:						
1% *Cr*, ½% *Mo*	A335, P12	15,000	14,750	13,100	7,500	2,800
1¼% *Cr*, ½% *Mo*	A335, P11	15,000	15,000	13,100	7,800	4,000
3% *Cr*, 1% *Mo*	A335, P21	15,000	13,900	12,000	7,000	4,000
18% *Cr*, 8% *Ni*	A312, Tp317	......	16,750	16,000	14,000	10,400

[a] Adapted from ASA B31.1-1955 and ASME 1956 Power Boiler Code.

TABLE 16-4. DIMENSIONS OF WELDED AND SEAMLESS STEEL PIPE[a]

Nominal pipe size	OD	Nominal wall thicknesses for schedule numbers										Nominal pipe size
		10	20	30	40	60	80	100	120	140	160	
⅛	0.405	....	....	....	**0.068**	....	*0.095*	....	....	....	....	⅛
¼	0.540	....	....	....	**0.088**	....	*0.119*	....	....	....	....	¼
⅜	0.675	....	....	....	**0.091**	....	*0.126*	....	....	....	....	⅜
½	0.840	....	....	....	**0.109**	....	*0.147*	....	....	....	0.187	½
¾	1.050	....	....	....	**0.113**	....	*0.154*	....	....	....	0.218	¾
1	1.315	....	....	....	**0.133**	....	*0.179*	....	....	....	0.250	1
1¼	1.660	....	....	....	**0.140**	....	*0.191*	....	....	....	0.250	1¼
1½	1.900	....	....	....	**0.145**	....	*0.200*	....	....	....	0.281	1½
2	2.375	....	....	....	**0.154**	....	*0.218*	....	....	....	0.343	2
2½	2.875	....	....	....	**0.203**	....	*0.276*	....	....	....	0.375	2½
3	3.500	....	....	....	**0.216**	....	*0.300*	....	....	....	0.437	3
3½	4.000	....	....	....	**0.226**	....	*0.318*	....	....	....	....	3½
4	4.500	....	....	....	**0.237**	....	*0.337*	....	0.437	....	0.531	4
5	5.563	....	....	....	**0.258**	....	*0.375*	....	0.500	....	0.625	5
6	6.625	....	....	....	**0.280**	....	*0.432*	....	0.562	....	0.718	6
8	8.625	....	0.250	**0.277**	**0.322**	0.406	*0.500*	0.593	0.718	0.812	0.906	8
10	10.75	....	0.250	**0.307**	**0.365**	*0.500*	0.593	0.718	0.843	1.000	1.125	10
12	12.75	....	0.250	**0.330**	0.406	0.562	0.687	0.843	1.000	1.125	1.312	12
14 OD	14.00	0.250	0.312	**0.375**	0.437	0.593	0.750	0.937	1.093	1.250	1.406	14 OD
16 OD	16.00	0.250	0.312	**0.375**	*0.500*	0.656	0.843	1.031	1.218	1.437	1.593	16 OD
18 OD	18.00	0.250	0.312	0.437	0.562	0.750	0.937	1.156	1.375	1.562	1.781	18 OD
20 OD	20.00	0.250	**0.375**	*0.500*	0.593	0.812	1.031	1.281	1.500	1.750	1.968	20 OD
24 OD	24.00	0.250	**0.375**	0.562	0.687	0.968	1.218	1.531	1.812	2.062	2.343	24 OD

Thicknesses shown boldface denote **standard-weight** pipe; those in italics *extra-strong* pipe. All dimensions given in inches.

[a] Adapted from "Navco Piping Datalog," 8th ed., National Valve and Manufacturing Company.

$$SN = \frac{1{,}000 \times P_i}{S} \tag{16-2}$$

where SN = schedule number

P_i = internal pressure, psig

S = allowable stress, psi

Instead of schedule number the terms *standard, extra strong, strong,* **etc.**, may be used as in Table 16-4.

16-4. High-temperature Effects on Piping. High-temperature steam in piping has several effects: it (1) decreases allowable working stress as in Table 16-2, (2) accelerates oxidation and corrosion, (3) produces expansion, and (4) makes pipe material creep. Expansion stresses the pipe supports and connected equipment. Thermal expansion can be found by

$$L_t = L_0 \left[1 + a \frac{t - 32}{1{,}000} + b \frac{(t - 32)^2}{1{,}000} \right] \tag{16-3}$$

where L_t = length at t F

L_0 = length at 32 F

a, b = constants for materials, as below in the following table:

Metal	a	b
Cast iron	0.00544	0.00174
Steel	0.00621	0.00162
Copper	0.00927	0.00124

To offset expansion stresses, piping systems use sliding expansion joints, expansion bends, normal bends, and corrugated sections. Expansion joints use two concentric nested sections with a packing gland between to prevent leakage. Most designers prefer expansion bends or a number of normal bends to sliding expansion joints (Fig. 16-1). Corrugated sections and bends are used for moderate pressures.

Under high temperature, many metals when stressed, will slowly deform in a plastic fashion to ultimate failure; this process is called *creep*. A stress when first applied to a hot metal will elongate it in elastic fashion. As the stress continues acting, the elongation will first grow at a diminishing rate; this is the first stage of creep. In the second stage the elongation grows at a uniform rate. In the third stage the elongation rate accelerates until the metal fails. Figure 16-2 shows the creep characteristics of a specific metal at 1200 F. Materials in high-temperature service must be carefully watched and their service time logged. As the end of third-stage creep approaches, the equipment must be replaced.

16-5. Pipe Supports. Brackets attached to building columns and hangers suspended from beams usually support piping in power plants. Anchor supports do not allow a pipe to move at that point, or they can move only in one direction or plane. Hangers may be a rigid type attached to beams or may be spring-supported. The latter allows limited pipe motion in any direction. Roller supports allow considerable motion along the pipe axis.

Sloping steam lines makes it easy to drain moisture to traps at the low points. Valves and tap lines must be provided to empty pipelines in removing them from service.

16-6. Valves. Valves control the fluid flow in pipelines. Valves have a cast *body* containing an opening, called a *seat,* through which the fluid flows. A *disk* fastened to a *stem* held in a *bonnet* covers the seat to shut off flow. The bonnet fastens to the valve body, which has two openings attached to the pipeline it controls, one opening on each side of the seat. Varying the opening between seat and disk controls the rate of flow.

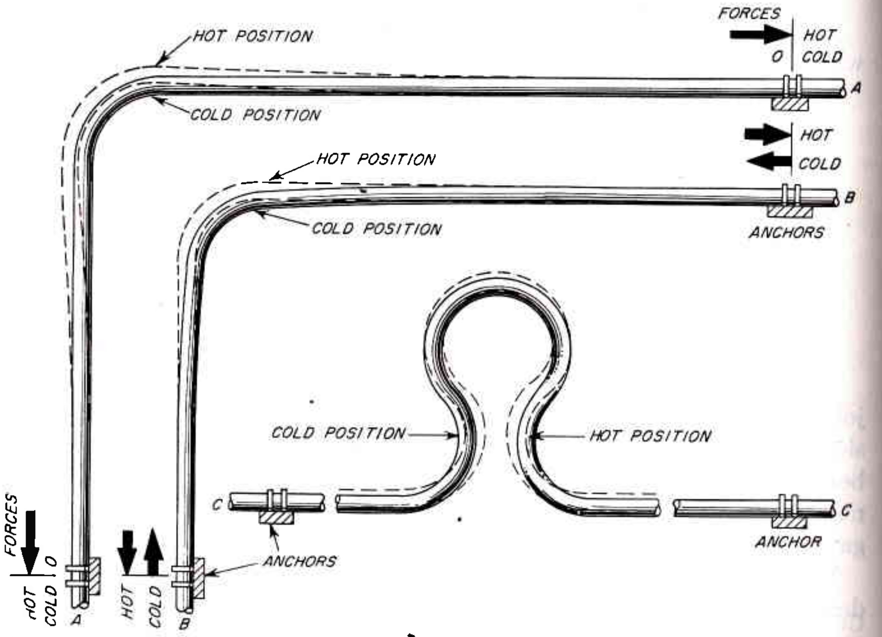


FIG. 16-1. *A* and *B* indicate how right-angle bends act when pipe expands while heating. In *B* the pipe has been cut short and put under tensile stress when cold, a condition called *cold springing.* The stress when hot is considerably less than *A.* A double-offset expansion bend as in *C* shows the effect when pipe is heated to relieve stress.

Figure 16-3 shows two examples each of *globe* and *gate valves.* Globe valves are used to control flow rate by throttling and to shut or stop flow completely. The flow area varies directly with the number of turns of the handwheel. Gate valves usually completely stop flow, normally being in either the full-open or completely closed position. They offer less flow resistance than a globe valve when completely open.

Power plants also use *check valves,* which have a flap allowing flow in one direction only; they close automatically when flow tries to reverse. *Needle valves* give micrometric flow control in small lines. *Safety,* or

relief, valves open to atmosphere when pressure in a line exceeds safe limits. Special valves include balanced valves for automatic flow regulation and seatless valves for blowdown lines.

Valves may have flanged, screwed, or welded ends and may be made of cast iron, cast steel, forged-steel alloys, brass, or other materials depending on the service conditions.

16-7. Traps. These automatic valves pass unwanted condensation in steam lines to waste but prevent steam from passing. They also work with steam-using equipment to drain off any accumulated condensation. Some traps also remove air and drain water from compressed air or gas lines. Traps are of two types: (1) nonreturn and (2) return. Nonreturn traps only separate, and discharge at a lower pressure than the inlet. Return traps lift or pump the fluid, discharging at a pressure equal to or greater than inlet pressure. Nonreturn traps have many arrangements: (1) open float, or bucket; (2) inverted bucket; (3) tilting; (4) ball float; (5) orifice; (6) impulse; and (7) thermostatic.

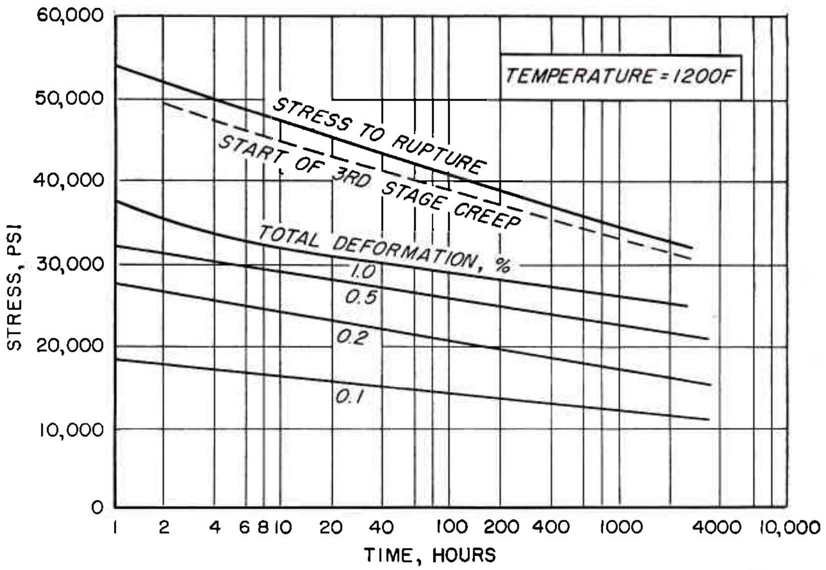
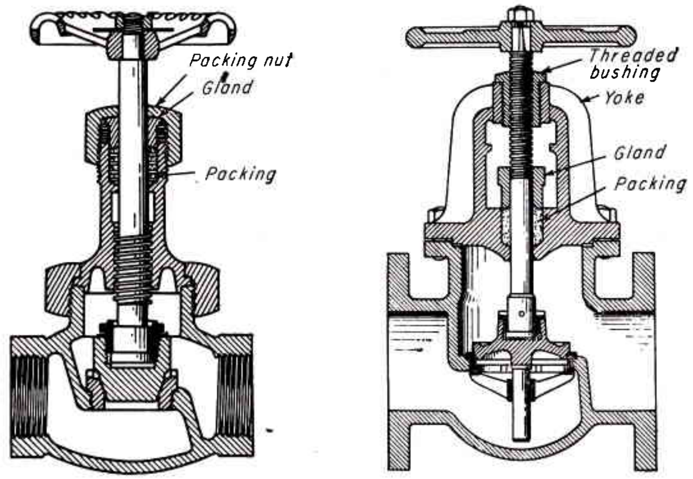


FIG. 16-2. Typical behavior of creep effect of a metal at one temperature. Designers use this type of data for a given life and operating temperature.

Return traps usually feed small amounts of condensate to a low-pressure boiler but may be used wherever condensate must be lifted a distance above the discharge point. All return traps have a receiver into which the condensate flows by gravity. Flow shuts off when the receiver fills, the vent closes, and boiler steam entering the tank forces the condensate out to a pressurized region. The process repeats automatically.

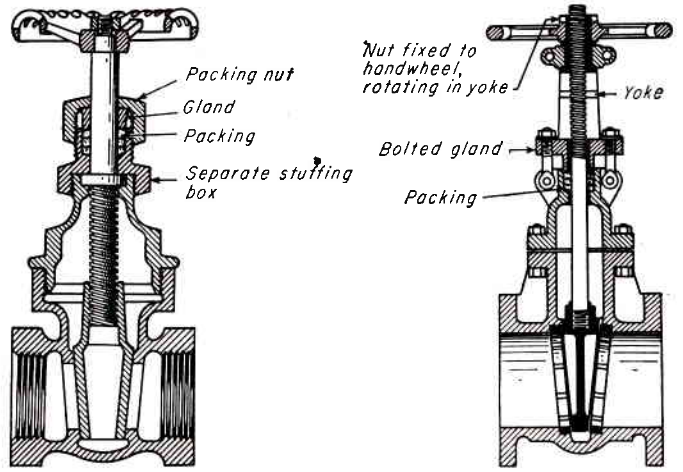


Inside screw, rising stem

(a)

Outside screw and yoke, rising handwheel

(b)



Inside screw, non-rising stem

(c)

Outside screw and yoke, non-rising handwheel

(d)

FIG. 16-3. Examples of common valve types. (*a*) Small inside screw, rising stem globe valve; (*b*) large outside screw and yoke, rising handwheel globe valve; (*c*) small inside screw, nonrising stem gate valve; (*d*) large outside screw and yoke, nonrising handwheel gate valve.

16-8. Pipe Fittings and Joints. Piping comes in standard-length sections. A piping system consists of sections joined together directly or through fittings and valves. Fittings include elbows, tees, reducing connections, crosses, plugs, unions, couplings, bushings, and blind flanges.

Methods of joining pipe sections, valves, and fittings depend on pipe size, pressure, and temperature of the fluid and pipe material. Small light-duty lines may use *threaded* or *soldered* connections. Medium and large lines may be *flanged* for moderate pressures. Flanges may be: (1) cast integrally with the pipe, (2) screwed on ends of threaded pipe, or (3) fitted with rings bearing on laps forged on pipe ends. High-pressure and -temperature service uses the third type. Flanges are bolted together at the contact faces, which may be serrated, ground and lapped, seal-welded, etc. For rough flange faces a gasket fills the gaps.

Present power plants largely use welded connections for the major steam and water lines to make connections tight. Welding makes pipe-insulation application easier. Welds must be stress-relieved and X-rayed or magna-fluxed for internal and surface inspections.

16-9. Pipe Insulation. Steam radiates energy through all surfaces it touches. Insulation applied to the exterior of pipe surfaces retards this loss and saves appreciable energy.

Useful materials are 85 per cent magnesia (85 per cent magnesium carbonate + 15 per cent asbestos fibers) and mineral wool (molten glass blown into fibers). Insulation may be applied in blocks, in formed shapes, or as a cement covering. A canvas covering over the insulation makes a good appearance.

High-temperature services may use up to 5-in. thicknesses of insulation. Thickness of insulation must be balanced against the cost of the energy saved. Initial piping design should allow for ease of applying insulation, especially in close quarters.

16-10. Piping Layouts. Piping layouts depend on the plant heat balance and must be individually designed. Code rules and standards set the minimum requirements that must be met. Ultimate designs are usually compromises between ideal layouts and over-all plant economics.

Figures 16-4 and 16-5 show the main features of steam and condensate piping in schematic form. Figures 16-6 and 16-7 show detailed schematics of steam and condensate piping systems and raw-water and air piping systems for a 35-mw unit using 900-psig 900-F steam. These systems anticipate the connections needed for a future new unit.

Industrial piping systems usually are more complex and varied than utility power-plant systems. Figure 16-8 shows schematically some of the additional types of equipment that industrial plants use. Some industries have their own complete power plants that work in conjunction with the equipment illustrated in Fig. 16-8.

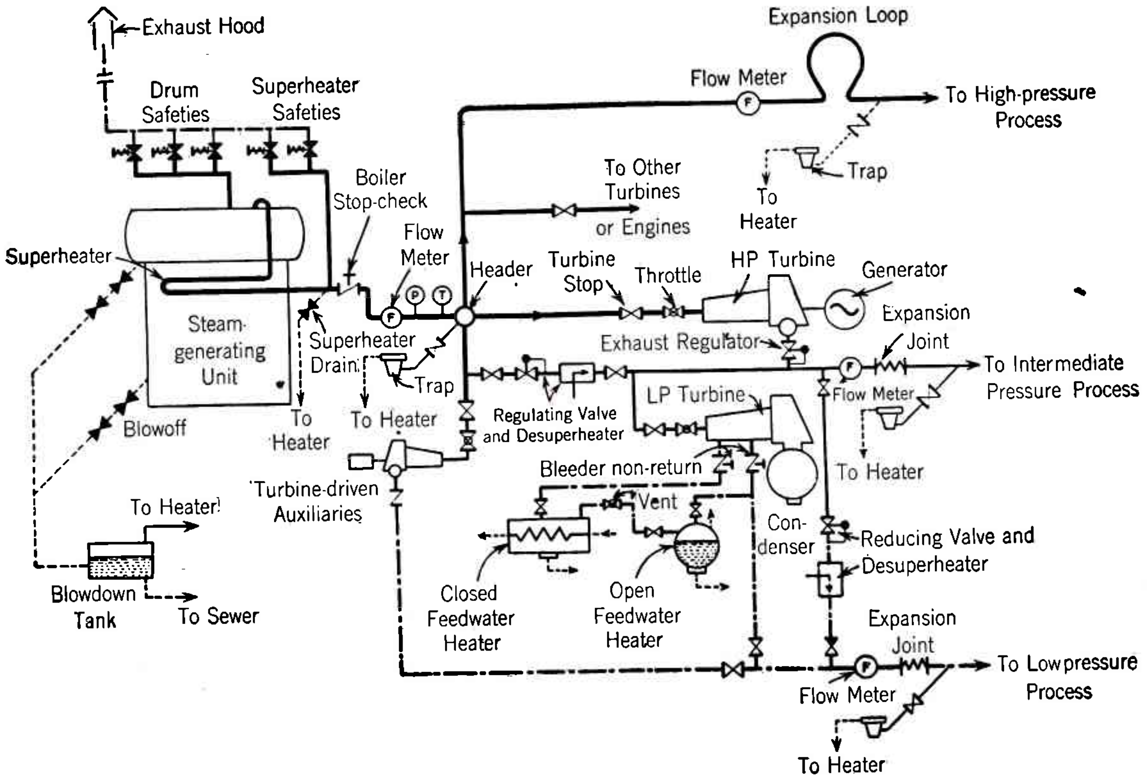


Fig. 16-4. Elements of power-station steam piping. (*Courtesy of Power.*)

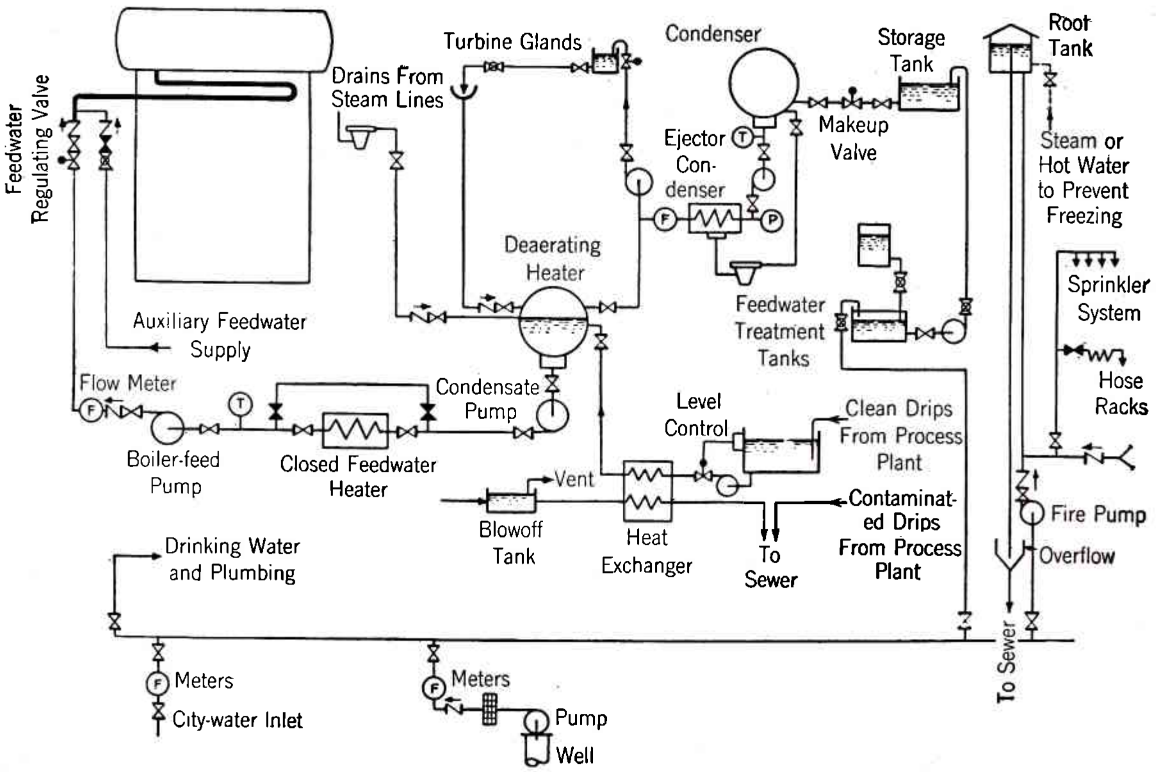


Fig. 16-5. Elements of power-station condensate and raw-water piping. (*Courtesy of Power.*)

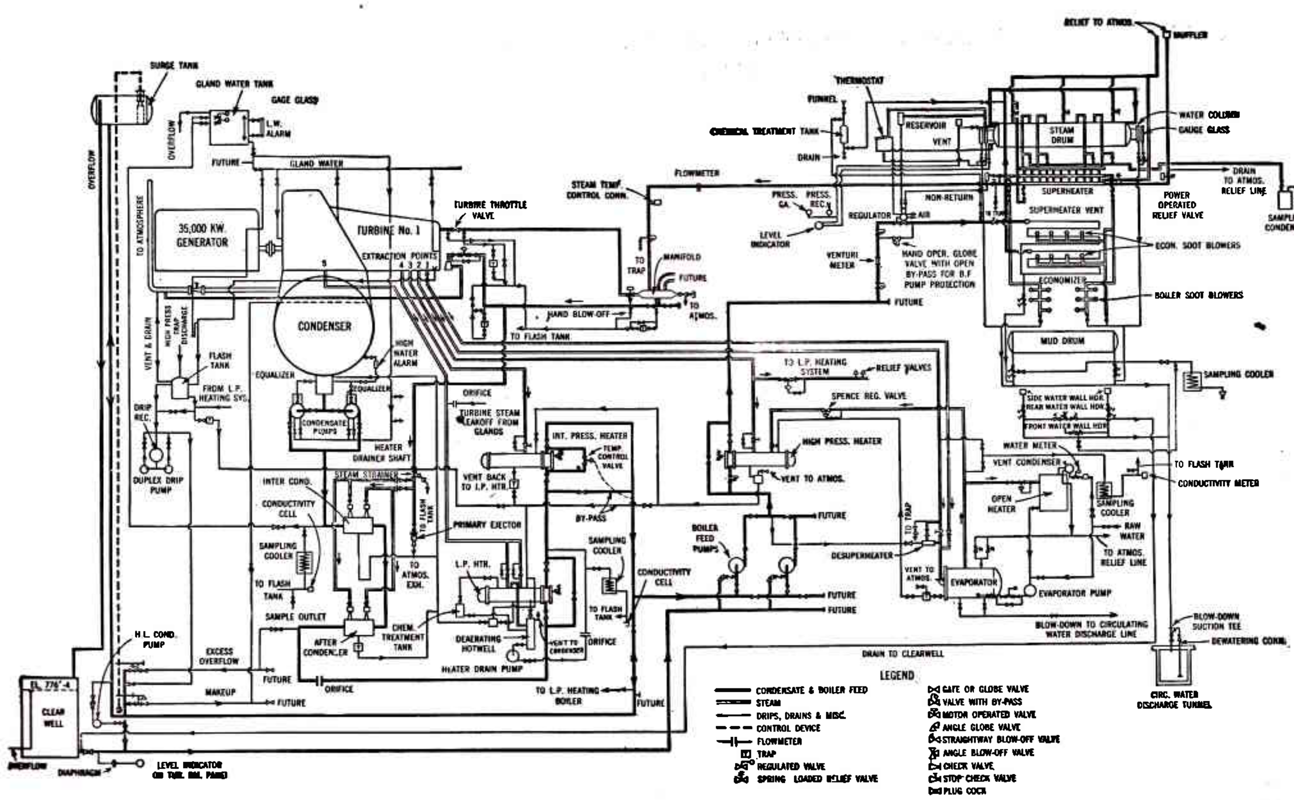


FIG. 16-6. Typical steam and condensate piping diagram for medium-capacity station. (*Courtesy of Commonwealth & Southern Corporation.*)

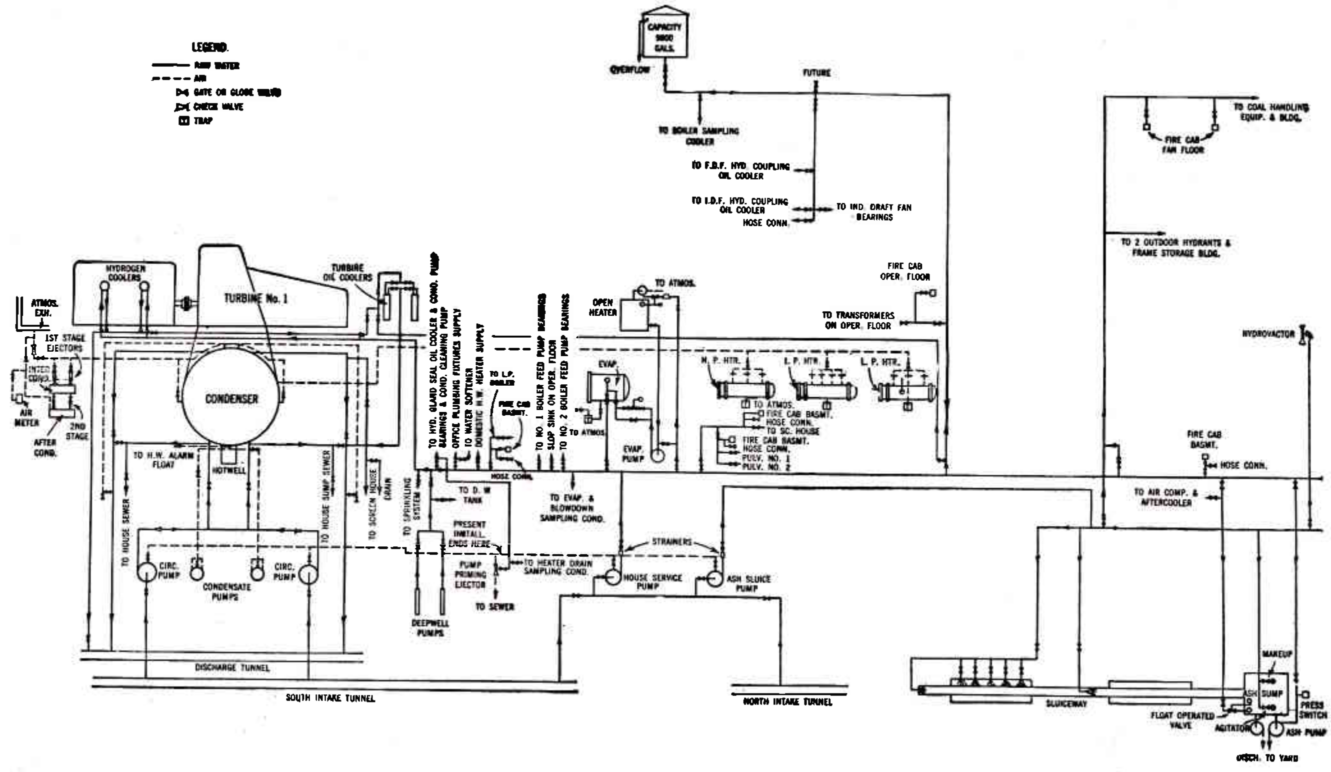


FIG. 16-7. Raw-water and vent piping for plant in Fig. 16-6. (*Courtesy of Commonwealth & Southern Corporation.*)

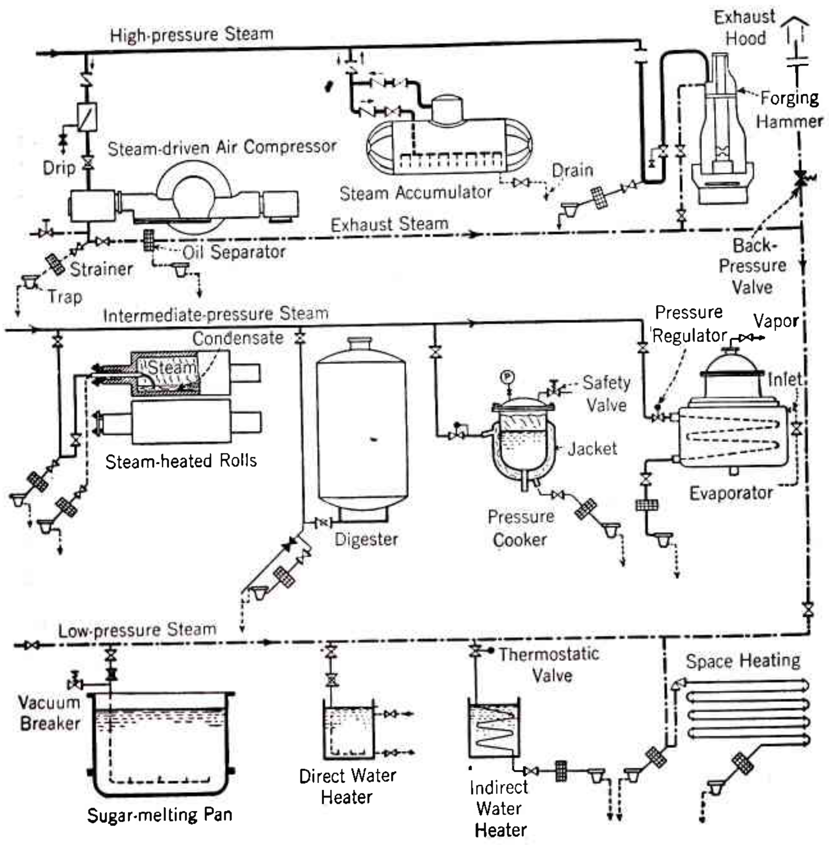


FIG. 16-8. Examples of process-plant steam uses and related piping. (*Courtesy of Power.*)

16-11. Steam-power-plant Heat Balance. A heat balance shows the computed mass and energy flows between the major equipment of a power plant, together with the pressures and temperatures. The flows are shown on a diagram of the cycle equipment arrangement. The computations enable finding the heat rate of the cycle. Figure 16-9 shows the heat balance for a typical low-pressure station.

Heat balances for an existing plant may be calculated from observed data on the actual equipment. Many heat balances are made for a projected plant and help determine the optimum arrangement.

Heat balances may be computed for several different loads, with diagrams for each load. Computations for large plants become complex, and systems must be developed to expedite the work. First draw a schematic diagram of the plant layout. This shows all major equipment and principal flow lines for steam, water, and gas; detailed piping and

valves are not essential. Equipment manufacturers furnish data for their products. For example, the turbine builder will supply a condition curve for various throttle flows. Similar performance characteristics will be available for the steam generator, feedwater heaters, evaporators, pumps, and fans. See Prob. 16-9 as an example of the data used.

Energy and mass flows of heaters, pumps, evaporators, turbines, and steam generators are each treated as separate items. The thermodynamic concept of a system and its boundaries is applied to each in turn. The first law applies to each item, assuming it a steady-flow device.

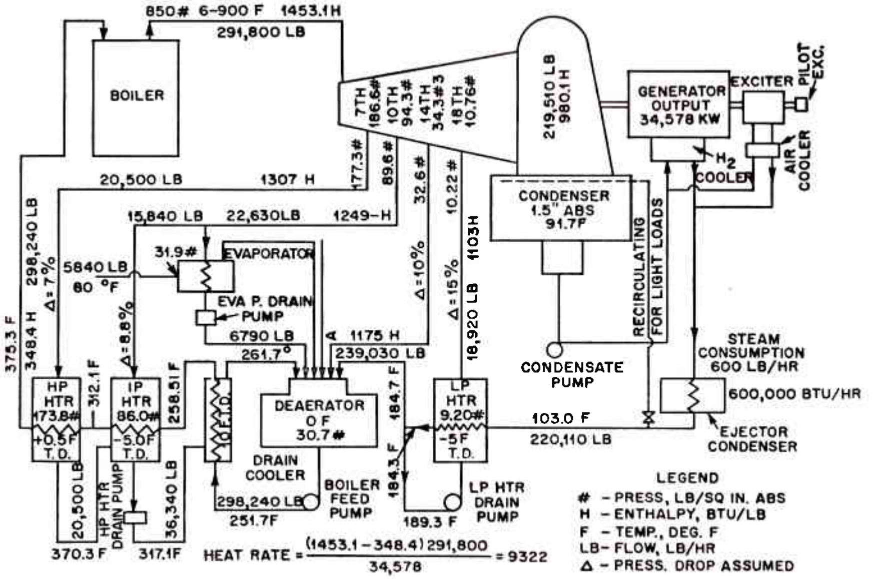
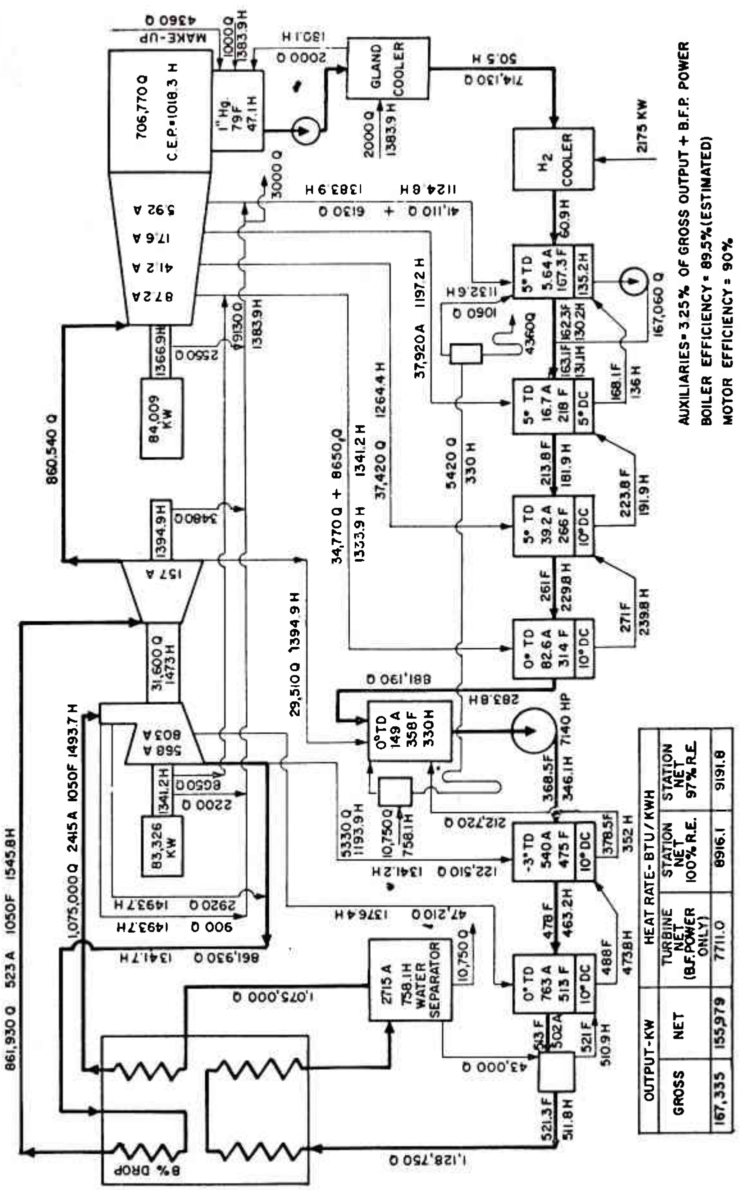


FIG. 16-9. Heat-balance diagram of Silver Gate Station, San Diego Gas and Electric Company, has 35-mw turbine operating at 850-psig 900-F 1.5-in.-Hg steam conditions. This heat balance is calculated for 34,578-kw output.

Calculations usually start at the high-pressure heater, evaporator, or other equipment where enough data are available to solve a First-Law equation. Calculations proceed to the lower-pressure heaters in turn, and the steam bled from the turbine at its several points may be found. Two unknowns in a balance can often be solved if the system is considered first as a single piece of equipment and then as two pieces in series.

When the bled and gland steam for the turbine is known, the work done by the steam on the turbine blades can be calculated. The *gross power* from the turbine is found by multiplying the blade work by a combined mechanical-electrical efficiency; subtracting auxiliary power gives *net power* for the cycle.

Heat energy to be supplied by the fuel is based on steam flow and


AUXILIARIES = 3.25% OF GROSS OUTPUT + B.F.P. POWER
BOILER EFFICIENCY = 89.5% (ESTIMATED)
MOTOR EFFICIENCY = 90%
8% DROP
WATER SEPARATOR
GLAND COOLER
H_2 COOLER
MAKE-UP

OUTPUT-KW		HEAT RATE-BTU/KWH		
GROSS	NET	TURBINE NET (B.F. POWER ONLY)	STATION NET 100% R.E.	STATION NET 97% R.E.
167,335	155,979	7711.0	8916.1	9191.8

Fig. 16-10. Heat-balance diagram of Portland Generating Station, Metropolitan Edison Company, has 150-mw turbine operating at 2,400-psig 1050/1050-F 1-in.-Hg steam conditions. Heat balance is calculated for 155,979 kw output.

boiler efficiency. The *net station heat rate* is

$$HR = \frac{\text{net station power}}{\text{heat energy in fuel}} \tag{16-4}$$

Figure 16-10 shows a heat balance for a reheat cycle using seven stages of feedwater heating.

PROBLEMS

16-1. Specify the trap capacity needed in pounds per hour for draining a closed feedwater heater if 40,000 lb of water per hour enters at 280 F and leaves at 325 F. Steam bled from the turbine is at 330 F and has 3 per cent moisture. The condensate is subcooled 10 F, and the radiation loss amounts to 5 per cent of the heat given to heat the feedwater.

16-2. A 4-in. (nominal) pipe carries steam at 1,200 psig and 900 F. Determine the schedule number for an alloy steel of Specification A355, P21.

16-3. A 5-in. (nominal) pipe carries 62,000 lb per hr of steam to a turbine using steam at 900 psia and 900 F. The pipeline is 110 ft long.

a. Designate a line with the necessary valves and fittings.

b. Calculate the boiler pressure required.

c. Specify a suitable pipe material.

d. Specify the pipe wall thickness and schedule number to be used.

16-4. Calculate Prob. 16-3 if steam is at 2,400 psi and 1000 F.

16-5. Calculate the heat loss in a 50-ft length of bare insulated pipe from the following data:

Pipe 3½ in. (nominal), schedule 80, horizontal
Temperatures, steam 650 F, outside air 70 F
Insulation, 85 per cent magnesia; surface emissivity 0.90, 2 in. thick

16-6. Investigate the most desirable thickness of insulation to be used in a piping system for which general data as follows are available. Select some specific set of values on which to base the solution.

Pipe, 8 in. (nominal), schedule 100, 200 ft horizontal
Temperatures, steam 900 F, outside air 80 F
Insulation, 85 per cent magnesia, surface emissivity 0.94
Steam-energy cost, 50 to 80 cents per million Btu
Cost of insulation installed as in Table P-1

Table P-1

Thickness, in.	*Dollars per ft*
1	0.80–1.10
2	1.25–1.75
3	2.50–3.00
4	3.50–4.50

Fixed charges, 20 to 25 per cent

16-7. Select a suitable material for the pipe in Prob. 16-4, and calculate the design pressure.

16-8. A steam line 220 ft long at 70 F is used to carry steam at 900 F. One end of the line is anchored, and the pipe rests on roller supports. Determine the expansion of the pipe.

16-9. a. Draw a line diagram of a regenerative power-plant hookup, using the equipment listed below and showing main lines and valves represented by standard symbols.

Indicate the following quantities in all lines in the order given when 70,000 lb per hr flows from the steam generator:

1. Absolute pressure, psia
2. Temperature, °F
3. Enthalpy values, Btu per pound
4. Mass flow, pounds per hour

b. Determine gross station heat rate for the foregoing condition for a steam-generator efficiency of 85.0 per cent and an electric-generator efficiency of 95 per cent.

MAJOR EQUIPMENT

1. One steam generator, 400 psia and 600 F steam.
2. One bleeder turbine rated at 6,000 kw with inlet steam throttled to 325 psia and exhausting to a condenser at 2 in. Hg back pressure; bleed pressures, 60, 18, and 4 psia; and operating at 3,600 rpm.
3. One closed heater, using 4 psia bled steam and receiving condensate from condenser by means of a motor-driven condensate pump; feedwater discharged to vent condenser; heater drained by motor-driven pump to discharge side.
4. One closed heater, using 60 psia bled steam receiving feedwater from boiler feed pump and discharging to boiler; condensed steam trapped to deaerating heater.
5. One deaerating heater and vent condenser, using 18 psia bled steam and exhaust steam from turbine driving feed pump. Heater *receives* feedwater from vent condenser, vent-condenser drains, and condensate from high-pressure heater. Heater *discharges* to boiler feed pump and vents steam and air to vent condenser.
6. One turbine-driven boiler feed pump receives feedwater from deaerating heater and discharges to high-pressure closed heater. Turbine uses steam from main line reduced to 300 psia by a pressure-reducing valve and discharges at 18 psia to deaerating heater.
7. One surge tank, in line from low-pressure heater to vent condenser.

DATA RELATIVE TO EQUIPMENT AND HOOKUP

1. Turbine condition curve data as in Table P-2.

TABLE P-2

Pressure abs	*Temperature or quality*
200 psi	500 F
55 psi	300 F
12 psi	4.7 per cent
2 psi	9.2 per cent
2 in. Hg	10.7 per cent

2. Boiler-feed-pump turbine data as in Table P-3.

TABLE P-3

Horsepower output	*Engine efficiency (based on hp output)*
8	45.0
20	47.9
35	50.0
65	52.5
100	54.0
150	55.0

3. Closed-heater and vent-condenser condensate leaves subcooled 5 F below saturation temperature for the pressure; feedwater leaves closed heaters 10 F below steam temperature.
4. Steam vented to vent condenser from deaerating heater equals 10 per cent of steam bled at 18 lb from turbine; 10 per cent of vented steam is discharged with the air.
5. Vent condenser raises feedwater temperature 5 F.
6. Consider no line pressure drops.
7. Boiler-feed-pump efficiency is 56 per cent at given conditions.
8. Neglect mechanical losses of boiler-feed-pump turbine.
9. Condensate pumps motor-driven with a negligible effect on enthalpy values.
10. Make-up to deaerating heater at 212 F equals steam vented to atmosphere.
11. Feedwater leaves saturated from deaerating heater and from main condenser.

16-10. a. Draw a line diagram showing main lines and valves for a regenerative power-plant cycle using the equipment listed below, represented by standard symbols.

b. Indicate, in so far as the data and calculations permit, the following quantities in all lines in the order given below. Show values for one or two loads as assigned.

1. Absolute pressure, psia
2. Temperature, °F
3. Enthalpy values, Btu per pound
4. Mass flow, pounds per hour

c. Determine the following for each load:

1. Gross plant generation, kilowatts
2. Net plant output, kilowatts
3. Net plant heat rate, Btu per kilowatt hour

EQUIPMENT DATA

1. Two steam generators, 1,300 psia, and 925 F steam.
2. One 1,800-rpm bleeder turbine rated at 100,000 kw with extraction points at 1, 3, 6, 9, 12, and 14 stages. Exhaust to a condenser pressure of 1.0 in. Hg abs.
   - A. *Calculate a plant heat balance for a throttle flow of* 546,000 *lb per hr* based on the stage data of Table P-4.
   - B. *Calculate a plant heat balance for a throttle flow of* 873,000 *lb per hr;* assume stage pressures vary directly as throttle flow to estimate stage pressures.
3. Major pump locations—all pumps motor-driven centrifugal.
   - *a.* Two boiler feed pumps in parallel—suction from No. 5 heater.
   - *b.* Two tank pumps in parallel—between heaters 3 and 4.
   - *c.* Two main condensate pumps in parallel, between main condenser and steam-jet air-ejector condenser.

Table P-4

Stage No.	*Data*
1	341 psia; no extraction economically warranted
3	211 psia; steam leads to steam-jet air ejector and No. 5 closed heater
6	101.0 psia; steam combines with turbine gland leakage and leads to No. 4 closed heater
9	41.1 psia; steam leads to No. 3 deaerating heater
12	14.5 psia; steam leads to No. 2 closed heater
14	4.3 psia; steam leads to No. 1 closed heater

4. Heater equipment flows.
   - *a.* No. 5 heater condensate flashed to No. 4 heater.
   - *b.* No. 4 heater receives boiler-feed-pump gland leakage; condensate flashed to No. 3 heater; No. 4 heater receives turbine high-pressure gland leakage.
   - *c.* No. 2 heater condensate flashed to No. 1 heater.
   - *d.* No. 1 heater condensate pumped into feedwater discharge line from No. 1 heater.
   - *e.* Air-ejector-condenser condensate flashed to main condenser hotwell; feedwater from air-ejector condenser flows to No. 1 heater.
5. Miscellaneous data and notes.
   - *a.* Make-up—2¼ per cent of throttle flow introduced at main condenser hotwell with h_f = 47.1 Btu per lb and atmospheric pressure.
   - *b.* Boiler blowoff, two-thirds of make-up.
   - *c.* Steam losses, one-third of make-up; consider as lost from extraction line to No. 5 heater.
   - *d.* Steam-jet air-ejector steam consumption 2,200 lb per hr for all throttle flows.
   - *e.* Pressure drop from turbine shell to heaters 10 per cent of extraction pressure.
   - *f.* Boiler feed pump and tank-pump motor and transformer efficiency 93 per cent for both station loads.
   - *g.* Calculate temperature rise through feedwater pump and tank pump, assuming it equal in effect to the inefficiency of the pump.
   - *h.* Consider the main condensate pump as having negligible effect on the enthalpy and temperature of water passing through it.
   - *i.* Main-condenser hotwell receives 110-F water from steam-jet air-ejector condenser and discharges saturated water at main-condenser pressure to condensate pump suction.
   - *j.* Total boiler-feedwater-pump gland leakage 63,400 lb per hr and 58,000 lb per hr for pump discharges, corresponding to throttle flows of 546,000 and 873,000 lb per hr, respectively; consider enthalpy equal to pump discharge enthalpy.
   - *k.* Base turbine mechanical losses on empirical equation.
   - *l.* Consider a negligible increase in enthalpy through the condensate pump from heater No. 1.
   - *m.* Consider water pressure between tank-pump discharge and feed-pump suction as 600 psia average; neglect subcooling effect between tank-pump suction and main-condenser hotwell pump discharge.

Performance Data

1. State line or condition curve (Table P-5).

Table P-5

873,000 lb per hr		546,000 lb per hr	
h, Btu per lb	*s*, units per lb	*h*, Btu per lb	*s*, units per lb
1387.0 (at No. 1 stage)	1.624	1366.0 (at No. 1 stage)	1.658
1350.0	1.632	1350.0	1.661
1300.0	1.642	1300.0	1.670
1250.0	1.652	1250.0	1.681
1200	1.663	1200.0	1.692
1150	1.676	1150.0	1.703
1100	1.688	1100.0	1.716
1050	1.703	1050.0	1.730
1000	1.719	1000.0	1.747
950	1.741	950.8	1.769
939	1.748		

2. Stage pressure-flow chart; consider curves as straight line from flow and pressure given through the origin (Table P-6).

Table P-6

Stage No.	Stage shell pressure, psia	Flow through following stage, lb per hr
1	571.0	900,000
3	385.0	900,000
6	191.5	900,000
9	81.5	900,000
12	30.6	900,000
14	9.7	900,000

Note: Values for stages 1 and 3 to be read to 2-psi accuracy; 6 and 9 to 1-psi accuracy; 12 and 14 to 0.1-psi accuracy.

3. High-pressure turbine gland leakage, straight line from data of Table P-7.

Table P-7

Gland leakage, lb per hr	*Throttle flow, lb per hr*
14,000	800,000
4,500	300,000

4. Turbine leaving velocity and exhaust loss, curve from data of Table P-8.

TABLE P-8

Exhaust loss, Btu per lb	Equivalent condenser flow, lb per hr	Exhaust loss, Btu per lb	Equivalent condenser flow, lb per hr
25.2	100,000	11.4	360,000
15.6	140,000	14.5	420,000
12.6	170,000	18.0	480,000
10.9	200,000	20.3	520,000
9.8	240,000	23.9	580,000
9.6	280,000	27.7	640,000
10.2	320,000		

NOTE: The above data apply for exhaust conditions of 1 in. Hg abs and 11.9 per cent moisture. To use curve for other conditions, apply following formula

$$Q_E = Q_A \frac{v(1 - y)}{575}$$

where Q_E = equivalent flow, lb per hr
Q_A = actual condenser flow, lb per hr
v = dry saturated specific volume, cu ft per lb
y = fractional moisture content from the state line

5. Generator performance curve from data of Table P-9 for 100,000 kw, 1,800 rpm, 13,800 volts, 0.9 power factor; 111,111 kva.

TABLE P-9

Generator output, 1,000 kw	Generator losses, kw	Generator output, 1,000 kw	Generator losses, kw
0	490	60	665
10	505	70	715
20	522	80	775
30	550	90	840
40	580	100	910
50	620		

6. Boiler efficiency (Table P-10).

TABLE P-10

Steam output, lb per hr	Efficiency, per cent	Steam output, lb per hr	Efficiency, per cent
300,000	87.55	700,000	88.10
400,000	87.78	800,000	88.05
500,000	87.95	900,000	87.80
600,000	88.08	1,000,000	87.15

7. Heater terminal temperature difference; performance curves are straight lines through the origin (Table P-11).

TABLE P-11

Heater No.	Water flow through heater, 1,000 lb per hr	Terminal-temperature difference, °F
5	1,200	6.4
4	1,100	4.7
2	900	4.0
1	700	4.2

NOTE: Condensate from all heaters is saturated at the heater pressure.

8. Data for all auxiliary power exclusive of boiler-feed-pump and tank-pump requirements; curve is straight line between points (Table P-12).

TABLE P-12

Generator load, kw	*Auxiliary power, kw*
100,000	4,000
10,000	2,200

9. Boiler feed pumps (Table P-13).

TABLE P-13

Flow, gpm	Total head, ft	Efficiency, per cent
0	3,170	0
200	3,170	26.5
400	3,165	44.0
600	3,150	57.0
800	3,075	66.0
1,000	2,950	71.5
1,200	2,750	74.0
1,300	2,550	73.0

10. Tank pumps (Table P-14).

TABLE P-14

Flow, gpm	Total head, ft	Efficiency, per cent
0	1,940	0
200	1,930	26.0
400	1,915	46.0
600	1,885	59.2
800	1,835	69.0
1,000	1,745	75.0
1,150	1,615	76.0
1,200	1,545	75.8
1,300	1,250	75.0

NOTES ON HEAT-BALANCE SOLUTION

1. Draw a cycle-equipment hookup diagram on 8½- by 11-in. sheet as a guide for use during solution of entire problem.

2. Draw final-equipment hookup diagram, using standard symbols on 18- by 24-in. sheet.

3. State line curve plotted on Mollier chart full size. Use fine inked line.

4. Stage pressure-flow chart on 11- by 17-in. graph sheet. All other graphs on 8½- by 11-in. sheet. Use hard pencil and plot accurately.

5. **Insert** values in heater and turbine tabulation sheet (Table P-15) from available data and computations as the solution progresses.

TABLE P-15. HEATER AND TURBINE TABULATION SHEET

Stage No.	3	6	9	12	14
Heater No.	5	4	3	2	1
Stage pressure, psia					
Heater pressure (0.9 × stage pressure), psia					
Saturated temp. in heater, °F					
Saturated h_f in heater, Btu per lb					
Water into heater, lb per hr					
Terminal temperature difference, °F					
Temperature feedwater out, °F					
h_f-feedwater out (uncorrected), Btu per lb					
Subcooled enthalpy correction, Btu per lb			Neg.	Neg.	Neg.
h_f-feedwater out (corrected), Btu per lb					
Bled-steam enthalpy, Btu per lb					
Bled steam, lb per hr					
Inlet steam to heater, lb per hr					
Flow past stage, lb per hr					

16-11. a. Draw a line diagram showing main lines and valves for a regenerative single-reheat power-plant cycle using the equipment listed below, represented by standard symbols.

b. Indicate the following quantities in so far as the data and calculations permit in all lines in the order given below. Show values for loads as assigned.

1. Absolute pressure, psia
2. Temperature, °F
3. Enthalpy values, Btu per pound
4. Mass flow, pounds per hour

c. Determine the following for each load:

1. Gross plant generation, kilowatts
2. Net plant output, kilowatts
3. Net plant heat rate, Btu per kilowatthour

MAJOR EQUIPMENT

1. One steam generator rated 1,800 psia, 1000 F superheater temperature, 1000 F reheat temperature.
2. One 3,600-rpm turbine generator rated at 156,250 kw with extraction points at six stages. Exhaust to a condenser pressure of 2.5 in. Hg abs.
   A. *Calculate a plant heat balance for a throttle flow of* 1,200,000 *lb per hr* with the evaporator in service. Base on stage pressure data as in Table P-16.

TABLE P-16

Stage No.	*Stage pressure, psia*
6(X-O)	554.4
9	230.5
Deaerator	132.9
13	73.7
15	28.5
17	10.2

   B. *Calculate a plant heat balance for a throttle flow of* 1,045,200 *lb per hr* with the evaporator out of service. Solution to be based on assumption of stage pressures and calculations to check the assumptions.

3. Basic plant layout information.

A power station in the southwest of the country is operated on a regenerative single-reheat steam cycle. The following equipment is used in the given order:

1. Boiler containing primary and reheat elements
2. Turbine, consisting of three shells mounted in tandem: high- and intermediate-pressure shells are of the single-flow type with high-pressure ends adjacent; they are followed by the double-flow low-pressure turbine (flow from the middle)
3. Condenser
4. Condenser pump
5. Gland steam exhauster (closed water heater)
6. 17th-stage heater (closed water heater)
7. 15th-stage heater (closed water heater)
8. 13th-stage heater (closed water heater)
9. Deaerator (direct mixing)
10. Boiler feed pump (two pumps in parallel)
11. 9th-stage heater (closed water heater)
12. Cross-over (X-O)
13. Evaporator (optional operation)

With the exception of the deaerator, in which feedwater and steam are mixed, the feedwater heaters have steam on the shell side and water on the tube side.

Steam flow occurs as follows in the given order:

A. From the high-pressure turbine.
   1. Gland steam from the high-pressure end to the intermediate-pressure turbine along the shaft.
   2. Gland steam from the low-pressure end is tapped from two points.
      a. From the point nearest to the shell to a bleeder line from the intermediate-pressure turbine for use in the 13th-stage heater.

  b. From the second point to a line carrying steam bled from the turbine inlet; one portion of this steam mixture is piped to the gland steam exhauster, and the remainder is combined with steam tapped from the low-pressure turbine for use in the 17th-stage heater.
  3. Turbine discharge returns to the boiler for reheating; the return line is tapped at two points: to the X-O heater, and to the evaporator (when in use).

B. From the boiler.
  1. Primary feed steam to the high-pressure turbine.
  2. Reheated steam to the intermediate-pressure turbine.
  3. Blowdown.

C. From the intermediate-pressure turbine.
  1. Bleed steam to the 9th-stage heater.
  2. Bleed steam to the deaerator.
  3. Bleed steam to 13th-stage heater (combined with a portion of the high-pressure-turbine gland steam).
  4. Discharge to the middle of the low-pressure turbine.
  5. Gland leakage may be neglected.

D. From the low-pressure turbine.
  1. Bleed steam from corresponding points on both sides to the 15th-stage heater.
  2. Bleed steam from corresponding points on both sides to the 17th-stage heater (combined with one part of the high-pressure-turbine gland-steam and inlet tap-steam mixture).
  3. Exhaust, combined from both sides, to the condenser.
  4. Gland steam may be neglected.

E. From the condenser feedwater goes to the condenser pump.

F. From the condenser pump to the gland-steam exhauster.

G. From the gland-steam exhauster.
  1. Feedwater to the 17th-stage heater.
  2. Condensate directly to the condenser hotwell.

H. From the 17th-stage heater.
  1. Feedwater to the 15th-stage heater.
  2. Condensate directly to the hotwell of the condenser.

I. From the 15th-stage heater.
  1. Feedwater to the 13th-stage heater.
  2. Condensate to the shell side of the 17th-stage heater.

J. From the 13th-stage heater.
  1. Feedwater to the deaerator.
  2. Condensate to the shell side of the 15th-stage heater.

K. From the deaerator feedwater goes to the boiler feed pump.

L. From the boiler feed pump (which consists of two units in parallel) it proceeds to the 9th-stage heater.

M. From the 9th-stage heater.
  1. Feedwater to the X-O heater.
  2. Condensate directly to the deaerator.

N. From the X-O heater.
  1. Feedwater directly to the boiler.
  2. Condensate to the shell side of the 9th-stage heater.

O. From the evaporator when in use.
  1. Make-up water (from shell side) to the deaerator.
  2. Condensate to the deaerator (separate line).

  3. Blowdown (from shell side).

Note that make-up water is added to the shell side of the evaporator. If the evaporator is inoperative, make-up water enters the hotwell of the condenser.

PERFORMANCE DATA

1. Gland leakage and enthalpy curves as in Table P-17.

TABLE P-17

Throttle flow, 1,000 lb per hr	Leakage No. 1, lb per hr	Enthalpy of leakage No. 1, Btu per lb	Leakage No. 2, lb per hr	Leakage No. 3, lb per hr	Leakage No. 4, lb per hr
300	4,800	1382	2,100	450	250
500	7,950	1399	3,500	750	400
700	11,150	1415	4,900	1,100	600
900	14,400	1431	6,300	1,400	750
1,200	19,200	1455	8,400	1,900	1,000

Leakage No. 1, first-stage packing leakage to intermediate-pressure turbine.

Leakage No. 2, first high-pressure-turbine exhaust leakage to 13th-stage heater (use high-pressure-turbine exhaust enthalpy).

Leakage No. 3, second high-pressure-turbine exhaust leakage to 17th-stage heater (use high-pressure-turbine exhaust enthalpy).

Leakage No. 4, valve-stem leakage to 17th-stage heater (use throttle enthalpy for leakage No. 4).

2. Extraction-stage-shell pressure curves (Tables P-18 to P-23). FFS = flow to following stage = throttle flow minus leakages and all extractions from preceding stages and stage in question, plus any steam returned to the turbine ahead of the stage in question. ESSP = extraction-stage-shell pressures.

TABLE P-18. X-O EXTRACTION STAGE

FFS, lb per hr	*ESSP, psia*
100,000	51
400,000	202
700,000	356
1,000,000	507

TABLE P-19. NINTH-STAGE EXTRACTION

FFS, lb per hr	*ESSP, psia*
100,000	23
400,000	90
700,000	156
1,000,000	223

TABLE P-20. DEAERATOR EXTRACTION STAGE

FFS, lb per hr	*ESSP, psia*
100,000	14
400,000	53
700,000	93
1,000,000	133

TABLE P-21. THIRTEENTH-STAGE EXTRACTION

FFS, lb per hr	*ESSP, psia*
100,000	8
400,000	31
700,000	54
1,000,000	77

Table P-22. Fifteenth-stage Extraction

FFS, lb per hr	*ESSP, psia*
100,000	3.2
400,000	12.7
700,000	22.1
1,000,000	31.5

Table P-23. Seventeenth-stage Extraction

FFS, lb per hr	*ESSP, psia*
100,000	1.3
400,000	4.9
700,000	8.5
1,000,000	12.1

3. State line or condition curves.

A. At 1,200,000 lb per hr throttle flow points for plotting expansion curve on Mollier chart as in Table P-24.

Table P-24

Pressure, psia	*Enthalpy, Btu per lb*
For intermediate- and low-pressure turbines:	
480	1518
250	1435
150	1378
100	1335
50	1270
20	1196
15	1175
10	1148
5	1106
2.5	1065
2.5 in. Hg	1028
For high-pressure turbine:	
First stage, 1,360	1455
End point, 554 psia	1353.3

B. At 1,045,200 lb per hr throttle flow points for plotting expansion curve on Mollier chart as in Table P-25.

Table P-25

Pressure, psia	*Enthalpy, Btu per lb*
For intermediate- and low-pressure turbines:	
430	1521
300	1475
200	1425
150	1392
100	1350
50	1282
26	1226
16	1190
5	1112
2.5	1074
2.5 in. Hg	1035
For high-pressure turbine:	
First stage, 1,140	1442
End point, 494	1342

4. Heater terminal temperature* difference curves. Plot on logarithmic paper (2 by 2 cycles).

Table P-26

Water flow, 1,000 lb per hr	°F terminal difference		
	X-O and 9th-stage heaters	13th-stage heater	15th- and 17th-stage heaters
150	0.148	0.118	0.104
300	0.600	0.468	0.417
600	2.42	1.90	1.68
1,200	9.65	7.50	6.75

5. Drain-cooler temperature curves as in Table P-27.

Table P-27

Drain flow 1,000 lb per hr	*Degrees of approach, F*[a]
X-O stage:	
20	2.7
70	9.5
110	14.9
140	19.1
9th stage:	
20	2.1
90	9.5
140	14.8
180	19.1
13th stage:	
10	2.6
40	10.5
60	15.9
70	18.5
15th stage:	
20	2.8
70	9.9
110	15.6
140	19.9
17th stage:	
20	1.6
80	6.7
140	11.8
190	16.0

[a] Degrees of approach = (temperature of bleed-steam condensate leaving heater) − (temperature of entering feedwater).

6. Turbine exhaust-loss curve. Exhaust loss at 1.0 in. Hg abs, 11.9 per cent moisture, 3,600 rpm (Table P-28).

---

* Terminal difference = (saturated-steam temperature within heater) − (temperature of leaving feedwater).

NOTES:

(1) To obtain the enthalpy of steam entering the condenser, multiply the exhaust loss by 0.96 (the approximate turbine internal efficiency), and add the result so obtained to the expansion-line end point.

(2) This exhaust-loss curve includes the loss in internal efficiency which occurs at light flows as obtained in tests.

(3) To use this curve for exhaust conditions other than specified above, read the exhaust loss at the equivalent flow obtained from the following expression,

$$Q_E = Q_A \frac{v(1 - y)}{575N}$$

where Q_E = equivalent flow per row of last-stage buckets, lb per hr
Q_A = actual condenser flow, lb per hr
v = saturated dry specific volume, cu ft per lb, corresponding to actual exhaust pressure. (when end point is in superheat region, v = actual specific volume at end point)
y = fractional moisture content at expansion-line end point
575 = specific volume, cu ft per lb, at 1 in. Hg abs, 11.9% moisture
N = number of rows of last-stage buckets (N = 2)

TABLE P-28

Equivalent flow per row of last-stage buckets (1,000 *lb per hr*)	*Exhaust loss, Btu per lb of condenser flow*
50	30.0
100	12.3
125	10.8
150	12.2
200	18.6
250	25.5
300	32.6
350	39.7
400	46.3
450	52.7
500	58.6
550	64.4
600	70.0

7. Boiler-feed-pump curve as in Table P-29.

TABLE P-29

Head, ft	Efficiency, %	Flow, U.S. gpm
6,180	11.5	100
6,170	21.5	200
6,150	39.0	400
6,100	52.0	600
6,050	61.0	800
5,950	68.0	1,000
5,800	73.5	1,200
5,650	77.5	1,400
5,400	79.0	1,600

8. Generator loss curve. Generator losses at 30 psig, H_2 pressure.

TABLE P-30

Loss, kw	*Net generator output, mw*
1,580	90
1,640	100
1,700	110
1,770	120
1,840	130
1,920	140
1,990	150
2,080	160
2,170	170
2,280	180

MISCELLANEOUS DATA AND NOTES

1. Assume 3 per cent pressure drop from *stage* to *stage flange* and 3 per cent pressure drop from *stage flange* to *heater*.
2. Assume 1 Btu per lb decrease in enthalpy as heat loss in steam lines from stage flange to heater.
3. Problem 17-6 gives data for computing evaporator flows at 1,200,000 lb per hr throttle flow.
4. Gland leakages 3 and 4 go to 17th-stage heater and gland steam exhauster. Gland exhauster takes 1,800 lb per hr at all throttle flows. Assume gland-exhauster shell-side pressure 13.0 psia and condensate leaves saturated at that pressure.
5. Assume negligible enthalpy increase in main-condenser condensate pump.
6. Base calculations on a steam-generator efficiency of 84.5 per cent, which is obtained with gas firing, for both throttle flows.
7. Auxiliary power data as in Table P-31.

TABLE P-31

Throttle flow, lb per hr	*Auxiliary power, kw*
1,200,000	6,970
1,045,000	6,540

8. Steam-generator performance as in Table P-32.

TABLE P-32

	1,200,000 lb per hr			1,045,000 lb per hr		
	p, psia	t, °F	w, lb per hr	p, psia	t, °F	w, lb per hr
Superheater outlet	1,925	1007	......	1,897	1005	
Drum conditions	2,006	636.2	......	1,964	633.2	
Reheater inlet	531.3	692.0	......	472.1	669	
Reheater outlet	499.4	1005	......	443.7	1005	
Blowdown	......	636.2	12,000	......	633.2	12,000

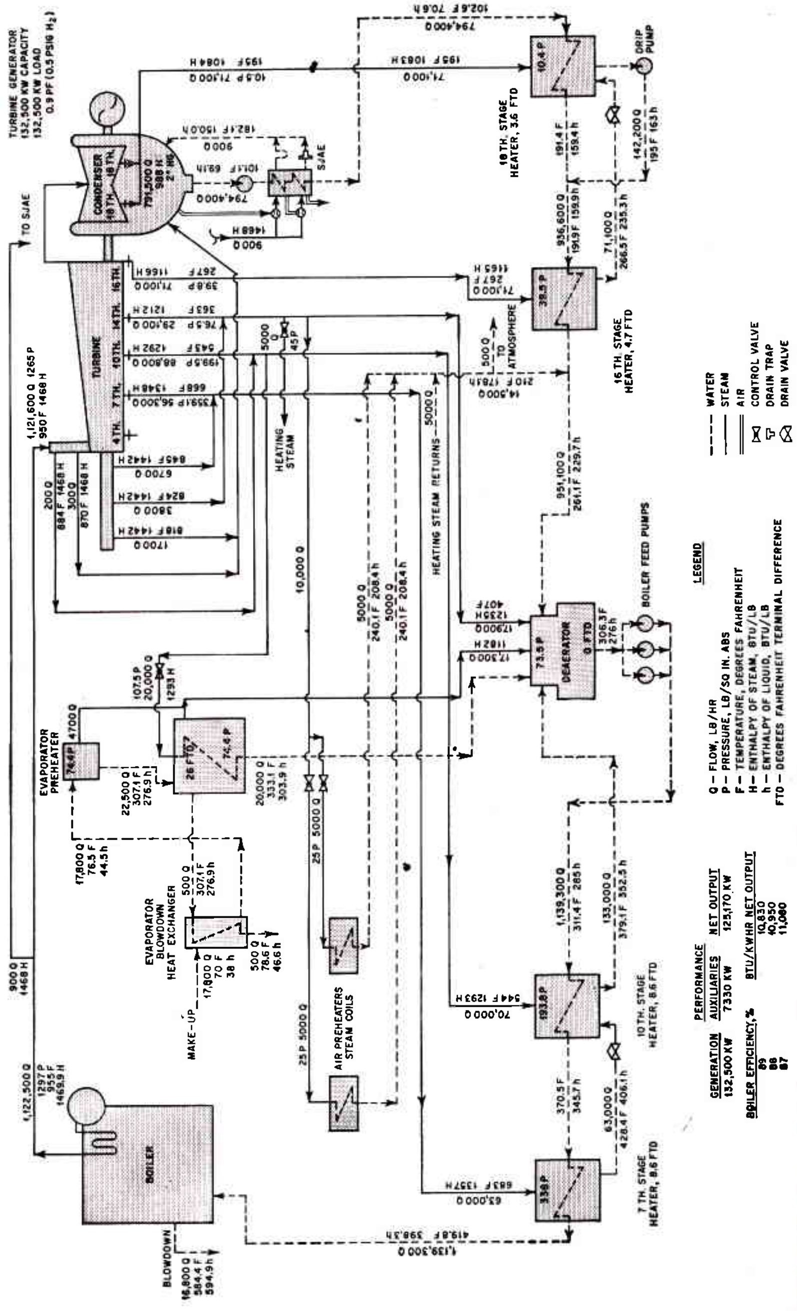


FIG. 16-11. Heat balance at full load for a 132.5-mw turbine running at 1,265-psia 950-F 2.0-in.-Hg steam conditions.

9. Base turbine mechanical losses on empirical data.
10. Air offtake from condenser goes to a mechanical evacuating pump and is not to be included in the heat balance.

16-12. Refer to the heat-balance diagram (Fig. 16-11), and determine or calculate the following items.

a. Check the gross turbine load by mass-flow–enthalpy calculations. Assume mechanical-electrical efficiency to be 97.2 per cent.

b. Determine the hourly coal supply to the steam generator in long tons per hour, based on a 13,690 Btu, per lb coal, with the steam generator operating at its highest reported efficiency.

c. Check the heat balance across the following heaters:

7th-stage heater
10th-stage heater
Deaerating heater
16th-stage heater
18th-stage heater

d. Determine the condenser cooling-water flow in gallons per minute based on an inlet temperature of 60 F and a terminal temperature difference of 5 F.

e. Calculate the gross turbine heat rate.

f. Calculate the plant net heat rate.

g. Check the following:

1. Heat balance of SJAE
2. Enthalpy of condensate from evaporator-blowdown heat exchanger.
3. Air flow over air-preheater steam coils in pounds per hour based on a 40 F rise in temperature of the air.

16-13. Refer to the heat-balance diagram of Fig. 16-10, and determine or calculate the following items:

a. Check the gross turbine-blade work load by mass-flow–enthalpy computations, and determine the mechanical-electrical efficiency of the unit.

b. Determine the coal supply to the steam generator in long terms per hour, based on a 13,690 Btu per lb coal. Note that the water separator receives wet steam from the monotube steam generator and delivers dry steam to the superheater.

c. Check the heat balance across the seven heaters.

d. Determine the condenser cooling-water flow in gallons per minute based on an inlet temperature of 60 F and a terminal temperature difference of 5 F.

e. Calculate the station net heat rate.

f. What percentage of the steam-generator heat transfer occurs in the monotube section, superheater, and reheater?

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CHAPTER 17

WATER CONDITIONING

17-1. Troubles Caused by Water. Solid and dissolved matter in water deposits out on heat-transfer surfaces in a power plant, especially in the steam generator. These materials, like dirt and scale, have low conductivities and impede heat transfer severely. In high-temperature surfaces of steam generators this reduction of heat flow raises the temperature of the metal and may cause it to fail if the deposit is on the water side. To prevent deposits or *surface fouling,* precautions must be taken to treat water before using it in power-plant apparatus.

When deposits do build up on a heat-transfer surface, it must be cleaned to maintain efficiency and prevent failure. Feedwater impurities may cause: (1) scale formation, (2) corrosion, (3) carry-over, and (4) embrittlement. An extensive technology of water treatment for feedwater, cooling water, and service water in power plants has developed over the years. This is largely the province of the chemist, but the power-plant designer must be familiar with the problems and their solutions.

17-2. Water Impurities and Treatment. Water is a world-wide traveler and takes many forms as it goes through its hydrological cycle. We tap this cycle to get our water from: (1) the surface—ocean, river, or lake, or (2) the ground—well or spring. At one time or another this water was rain and it picks up oxygen, nitrogen, and carbon dioxide along with dust, fumes, smoke, and other matter as it falls. Surface waters accumulate silt, organic matter, sewage, and industrial wastes. Ground water dissolves and carries with it mineral matter as salts of calcium and magnesium.

Water may be analyzed and reported in three ways:

1. *Ionic.* Ions shown as Ca, Mg, $Na(HCO_3)_2$, SO_4, Cl, etc., to include all impurities.
2. *Combination.* Compounds as $CaCO_3$, $CaSO_4$, $MgCO_3$, etc., to include all impurities.
3. *Hardness.* Based on standard soap test indicating total hardness as $CaCO_3$, calcium hardness as $CaCO_3$, etc., for all contents.

Modern practice gives all analyses as *parts per million* (ppm) of the item; older practice states them as *grains per gallon* (17.1 ppm = 1 grain per gal).

The ionic form is easy to determine and may be converted to the combination form by empirical rules. The ionic form gives the basic information needed to make treatment calculations. The hardness analysis, while not as precise as the ionic, can be made quickly and proves useful for operating control showing changes in water condition.

Water authorities give the specific water impurities and their control as:

Color. Causes foaming in boilers and may interfere with treatments; may be removed by coagulation and filtration, chlorination, or adsorption by activated carbon.

Algae and slime. Cause coating on cooling-tower surfaces and cooling-water side of heat exchanger and clog spray ponds; removed by chlorination.

Suspended solids. Consist of colloidal matter that deposits on equipment surfaces; removed by settling, coagulation, and filtration.

Dissolved solids. Cause foaming in boilers; removed or decreased by distillation, demineralization, or softening.

Oil. Causes sludge, scale, and foaming in boilers; removed by strainers, coagulation and filtration, and baffle separators.

Hardness. Indicates presence of calcium and magnesium salts which scale up pipelines, heat exchangers, and boilers; removed or decreased by distillation, softening, or internal boiler treatments.

Alkalinity. Indicates presence of sodium bicarbonate, carbonate, and hydroxide, which cause foaming in boilers, carry-over, and embrittlement of boiler steel; removed or decreased by lime and lime-soda softening, acid treatment, hydrogen zeolite softening, demineralization, distillation, or dealkalization by anion exchange.

Free mineral acid. Usually present as sulfuric or hydrochloric acid and causes corrosion; reduced by neutralization with alkalies.

Carbon dioxide. Causes corrosion in steam, water, and condensate lines; removed by aeration, deaeration, or neutralization with alkalies or filming and neutralizing amines.

Oxygen. Causes corrosion and pitting of water lines, boilers, return lines, and heat exchangers; removed by deaeration, sodium sulfite, corrosion inhibitors, or hydrazine.

Sulfate. Causes calcium sulfate scale by combining with calcium; removed by distillation, soda ash, phosphate, or demineralization.

Chloride. Causes increased corrosive action of water; removed by distillation or demineralization.

Silica. Causes very hard scale in boilers and cooling-water systems and forms insoluble deposits on turbine blades; removed by hot process with magnesium salts, distillation, or adsorption by highly basic anion-exchange resins together with demineralization.

Iron. Causes deposits in boilers, heat exchangers, and water lines; removed by aeration, cation exchange, lime softening, coagulation and filtration, or surface active reagents.

Ammonia. Causes corrosion of copper and zinc alloys; removed by cation exchange with hydrogen zeolite, deaeration, or chlorination.

pH *index.* Indicates the hydrogen-ion concentration in water; neutral water has a pH of 7.0; values less than 7.0 show acidity and greater than 7.0 show alkalinity. The pH of water can be varied by adding alkalies or acids. Natural waters range usually from 6 to 8 pH.

17-3. Removing Suspended Matter. Treatment processes usually remove suspended material before the dissolved matter:

Sedimentation. Involves allowing water to stand quietly or flow at very low velocity. The solid matter settles, to be removed periodically or continually from the bottom. Clear water drains from the surface of the settling chamber.

Coagulation of minute colloidal suspensions makes them settle out easily. Adding a coagulant like aluminum sulfate or sodium aluminate improves the sedimentation or filtration process. A reaction between these salts and alkalinity in the water forms a gelatinous material (floc). The floc makes small particles adhere to each other, forming larger particles that settle out or filter out more easily.

Filtration. Makes dirt-laden water flow through beds of material like graded sand or anthrafilt. The suspended matter adheres to the filter material, leaving the water clear as it drains from the bottom. The beds must be backwashed periodically to remove the dirt that collects in the voids of the filter material. Water flows by gravity through a gravity filter. Figure 17-1 shows a pressure filter that may be placed in a line fed by a pump.

17-4. Removing Dissolved Gas. Water containing dissolved CO_2, H_2S, and other gases corrodes metals. *Aeration* and *degasification* remove these gases before chemical treatment. In aeration water is sprayed through towers to give the water a maximum surface through which the gases may escape. Aeration also destroys organic matter in sewage-polluted water.

17-5. Preventing Scale, and Removing Sludge. The lime-soda softening process uses calcium hydroxide (lime) and sodium carbonate (soda ash) to remove dissolved calcium and magnesium salts, by precipitating them. This may be done by either a *cold process* or a *hot process.* The processes use these types of reactions:

$$Mg(HCO_3)_2 + 2Ca(OH)_2 = Mg(OH)_2\downarrow + 2CaCO_3\downarrow + 2H_2O \qquad (17\text{-}1)$$
$$MgCl_2 + Ca(OH)_2 = Mg(OH)_2\downarrow + CaCl_2 \qquad (17\text{-}2)$$
$$CaCl_2 + Na_2CO_3 = CaCO_3\downarrow + 2NaCl \qquad (17\text{-}3)$$
$$CaSO_4 + Na_2CO_3 = CaCO_3\downarrow + Na_2SO_4 \qquad (17\text{-}4)$$

The arrows show the molecule precipitated from the solution. In Eq. (17-1) all the carbonate hardness precipitates out, leaving pure water. In Eqs. (17-2) to (17-4) the noncarbonate hardness precipitates, to be replaced by soluble sodium salts. The magnesium hydroxide [$Mg(OH)_2$] acts as a coagulant and has the property of adsorbing soluble silica from solution.

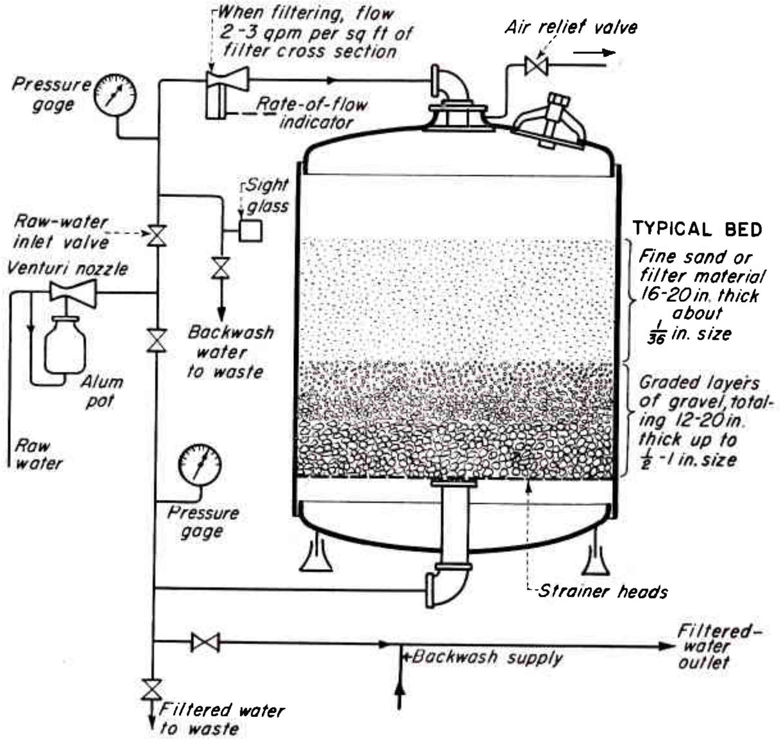

FIG. 17-1. Typical vertical shell pressure filter. Note the alum pot for addition of a coagulant and piping for *direct* and *backwash* operation.

Figure 17-2 shows a hot-process softener used in power plants. The original water hardness usually is not completely removed, because of the chemical cost. Even when some excess chemicals are used, the effluent hardness will range from 10 to 30 ppm.

Hot-process phosphate softening uses trisodium phosphate and caustic soda to precipitate and remove calcium and magnesium hardness. The reactions are

$$3Ca(HCO_3)_2 + 6NaOH = 3CaCO_3\downarrow + 3Na_2CO_3 + 6H_2O \qquad (17\text{-}5)$$
$$3CaCO_3 + 2Na_3PO_4 = Ca_3(PO_4)_2\downarrow + 2Na_2CO_3 \qquad (17\text{-}6)$$
$$Mg(HCO_3)_2 + 4NaOH = Mg(OH)_2\downarrow + 2Na_2CO_3 + 2H_2 \qquad (17\text{-}7)$$
$$3CaSO_4 + 2Na_3PO_4 = Ca_3(PO_4)_2\downarrow + 3Na_2SO_4 \qquad (17\text{-}8)$$

Equipment as in Fig. 17-2 carries out the operation at 212 F or higher. Phosphates also remove silica compounds, which are troublesome in high-pressure boilers. Phosphate chemicals cost more than lime soda, and so a two-stage process is used for waters with high hardness, 60 ppm and up. A first-stage lime-soda softener is followed by a second-stage phosphate treatment. The latter tends to produce high alkalinity and may be offset by adding acid phosphate (H_3PO_4) or sulfuric acid.

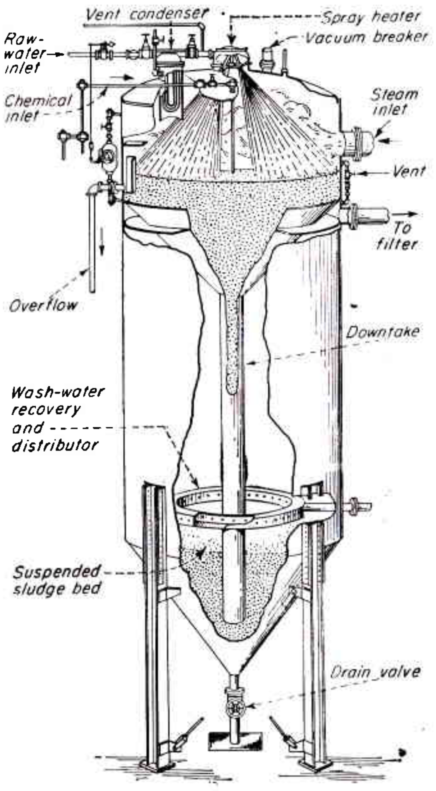

Fig. 17-2. Typical hot-process softener of sludge blanket design. Note water in downtake flows down, then up, through a bed of its own sludge before leaving for filtering. This action improves softening, reduces chemicals required, lowers turbidity of water, and reduces silica. Basically raw water enters and is heated by steam and mixed with a proportioned treatment of chemicals. Note that deaeration may be carried out in conjunction with the hot-process softening.

Silica removal became important with the advent of high pressures above 1,000 psig. Silica tends to form hard boiler deposits and deposits as a hard scale on certain turbine stages. It evaporates with the water at the higher pressures and condenses at lower pressures in the turbine as the steam expands. The hot-process lime soda and phosphate treatment removes part of the silica.

Magnesium compounds may be used to remove silica by adsorption, instead of by a chemical reaction. Magnesium compounds may be magnesium sulfate, dolomitric lime, calcined magnesite, magnesium carbonate, or magnesium oxide. An effective process uses 212 F, with retention times up to 1 hr and a pH of about 10, and recirculates the sludge. Silica may also be removed by a cold process using ferric iron compounds, such as ferric sulfate or hydrous ferric oxide, and the adsorption method. But the hot method is preferred because ferric salts increase the water solids content and a large amount is needed.

Sodium zeolite softening uses one form of zeolites, which are solid materials of complex chemical structure and of a sandy texture. They

remove various ions from the water, replacing them with other ions of like charge. In the usual sodium zeolite process the sodium of the softening bed replaces the calcium and magnesium that make up the hardness. This makes the bed change to calcium and magnesium zeolite, while the sodium forms of the salts leave with the treated water.

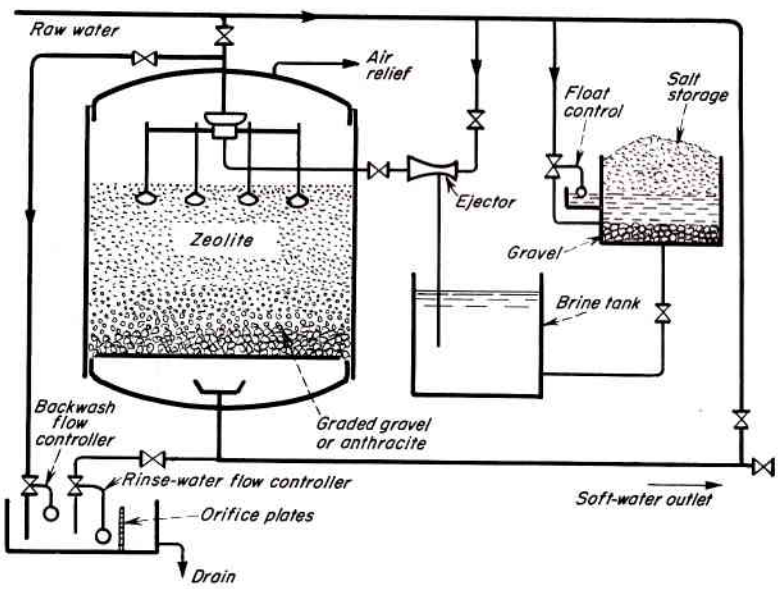

Fig. 17-3. Typical equipment for base-exchange zeolite softener. The zeolite is supported on a bed of graded sand and gravel or anthracite.

The reaction removes positive ions or cations from water,

$$Ca^{++} + Na_2Z = CaZ + 2Na^{++} \qquad (17\text{-}9)$$

When the bed no longer supplies sodium ions, it is regenerated with sodium chloride,

$$CaZ + 2NaCl = Na_2 + CaCl_2 \qquad (17\text{-}10)$$

Figure 17-3 shows a typical zeolite-softener diagram and resembles a filter as in Fig. 17-1, with necessary auxiliary piping for normal and regeneration processes. Zeolites almost completely remove hardness but do not reduce alkalinity or total solids. They are limited to about 140 F and below; too low or high a pH affects them. Turbid water does not react well with zeolite sands. They do not remove silica and may even add it. Effluent from a zeolite bed, being low in calcium, may be corrosive. A postacid treatment may reduce excess alkalinity from the sodium zeolite. Over-all treatments ordinarily use zeolite beds as one of the processes.

Hydrogen zeolite softening uses a nonsiliceous material that exchanges hydrogen ions for cations as calcium and magnesium when used in equipment similar to that of Fig. 17-2. Typical reactions are

$$Ca(HCO_3)_2 + H_2Z = CaZ + 2H_2CO_3 \qquad (17\text{-}11)$$
$$MgCl_2 + H_2Z = MgZ + 2HCl \qquad (17\text{-}12)$$
$$Na_2SO_4 + H_2Z = NaZ + H_2SO_4 \qquad (17\text{-}13)$$

Acids, mostly sulfuric, regenerate exhausted beds with the reaction

$$\left.\begin{matrix}Ca\\Mg\\Na_2\end{matrix}\right\} Z + H_2SO_4 = H_2Z + \left.\begin{matrix}Ca\\Mg\\Na_2\end{matrix}\right\} SO_4 \qquad (17\text{-}14)$$

Aeration removes the carbonic acid that decomposes after being produced according to Eq. (17-11). This removes the bicarbonate alkalinity without substituting other salts and so reduces the total solids. Zeolites are suitable for boiler water low in hardness and high in bicarbonate alkalinity. Running sodium and hydrogen zeolites in parallel and post treatment with alkali or dilution with raw water neutralizes the acidity of the effluent. Hydrogen zeolites cannot be used with turbid water and may be costly. The effluent of parallel degasified hydrogen-sodium zeolites may be corrosive because of dissolved oxygen.

Anion Exchangers. Anions such as chlorides, sulfates, and nitrates may be removed from water with certain resinous materials. These materials adsorb anions when the latter are in acid forms by removing the entire molecule. For example,

$$2HCl + RCO_3 = RCl_2 + H_2CO_3 \qquad (17\text{-}15)$$

Aeration removes the carbonic acid. A typical reaction for regenerating the anion exchanger is

$$RCl_2 + Na_2CO_3 = RCO_3 + 2NaCl \qquad (17\text{-}16)$$

These exchangers use the same type of equipment as the hydrogen and sodium zeolites.

Demineralization. Water may be freed of its mineral content by evaporation or by a series of cation and anion exchangers to produce essentially distilled water. Demineralizing often is the most economical method of producing make-up water for high-pressure boilers. Figure 17-4 shows a series process for demineralization. Raw water flows through a hydrogen zeolite exchanger (cation removal) and then to an anion exchanger, followed by aeration or degasification. The final exchanger adsorbs silica.

A simple demineralizer would use only the first two units. An expanding variety of ion-exchange material is being developed for demineralizing systems. Strong-base anion-exchange material adsorbs weak acids (silica types) as well as strong acids. Weak-base anion-exchange materials are most efficient for removing chlorides, sulfates, and nitrates. Sometimes mixed-bed anion-exchange materials are used to "polish" the effluent of the main anion exchangers.

Acid treatment of feedwater by direct addition of acid may be replaced by using the acidic effluent of the hydrogen zeolite process, usually in parallel with a sodium zeolite unit. Sulfuric acid is most often used for direct addition, but sometimes phosphoric or hydrochloric acid may be better. Acid treatments may be used to reduce alkalinity, increase the sulfate-carbonate ratio, reduce the carbon dioxide content of steam leaving a boiler, or prevent disintegrating the zeolite bed if the pH of the make-up is too high. Equipment for direct addition includes: acid-resistant tanks, proportioning feeders, and degasifiers or deaeration chambers for carbon dioxide removal.

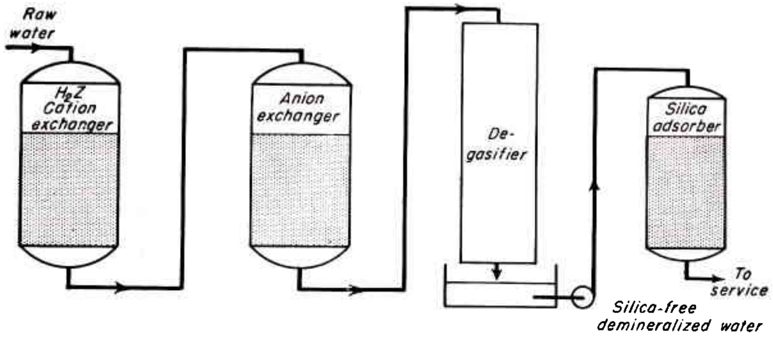

FIG. 17-4. Typical arrangement of demineralizing system using two exchangers, degasifier, and adsorber.

Evaporation by evaporators (Fig. 14-9) may be used to supply small amounts of make-up water in a closed cycle in place of chemical treatment. The feedwater to the evaporator, however, is usually treated. The evaporator steam (make-up) enters the main cycle by condensing in the deaerating heater. Economics governs the choice between chemical treatment and evaporation. Evaporators are limited to less than 5 per cent make-up.

17-6. Corrosion. Corrosion in power-plant equipment produces pits, grooves, and cracks or a general wastage of the wall material. Allowed to continue, corrosion ultimately makes metal parts fail. Corrosion mechanisms are under close study. While the results are not conclusive, the mechanisms appear to act as follows:

Dissolved gases are the most prevalent cause of corrosion; oxygen may be the most active, but CO_2, NH_4 and H_2S are also detrimental. Dissolved O_2 may enter in the make-up water or through leaks in the condensate system around the hotwell and low-pressure heaters.

Figure 17-5 outlines the action of dissolved O_2 in water in contact with iron; iron ions replace the hydrogen in ionized water. Hydrogen depositing on the iron tends to form a film, but this hydrogen combines with dissolved oxygen to expose the iron surface continuously. The iron ions react as follows:

$$Fe^{++} + 2OH^- = Fe(OH)_2 \qquad (17\text{-}17)$$

$$2Fe(OH)_2 + O_2 + 2H_2O = 2Fe(OH)_4 \qquad (17\text{-}18)$$

The material formed is ferric hydroxide $[Fe(OH)_4]$, or common rust; ferric oxide ($Fe_2O_3{\cdot}3H_2O$) may also form.

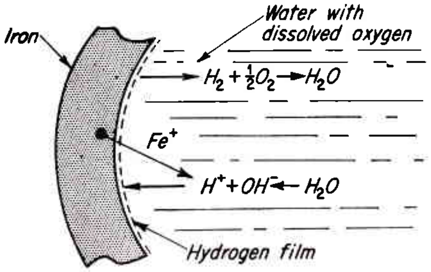
Iron
Water with dissolved oxygen
$H_2 + \frac{1}{2}O_2 \rightarrow H_2O$
Fe^+
$H^+ + OH^- \leftarrow H_2O$
Hydrogen film

FIG. 17-5. Effect of iron in contact with water-dissolved oxygen.

Corrosion from oxygen frequently appears in high-temperature regions as pits or depressions with a hard crust and having an underlayer of black magnetic oxide of iron (Fe_3O_4) or red ferric oxide (Fe_2O_3). Keeping oxygen out of the water prevents the dissolved-gas type of corrosion. This can be done *mechanically* with deaerating heaters as in Chap. 14 or *chemically* by treatment. The deaerator leaves traces of oxygen, and supplemental chemical treatment removes it. The treatment produces the reaction

$$2Na_2SO_3 + O_2 = 2Na_2SO_4 \qquad (17\text{-}19)$$

The sodium sulphate is soluble. A newer and promising method uses hydrazine with the reaction

$$N_2H_4 + O_2 = 2H_2O + N_2 \qquad (17\text{-}20)$$

Hydrazine solution is a clear, water-white, alkaline liquid with an odor like ammonia. It contributes alkalinity with end products of water and no solids. The nitrogen evolves with the steam.

Carbon dioxide may be in the make-up water or may be formed by decomposition of sodium bicarbonate in the water at boiler temperature. Corrosion by CO_2 appears as a general wastage or grooving of the metal in the steam space of a drum. The gas evolves with the steam. Boiler corrosion of this type is not common.

Corrosion from ammonia and hydrogen sulfide is not common either, because these gases are rarely found in feedwater. But ammonia tends to attack copper or its alloys, and hydrogen sulfide causes acidic corrosion similar to that caused by carbon dioxide.

Acidity. Since corrosion proceeds by metal ions replacing hydrogen ions, waters that are relatively acidic (low pH) tend to be corrosive. This appears as a general thinning and corrosion of the metal and happens frequently in economizers. Soda ash or sodium hydroxide is injected for alkalinization and keeps pH at about 10.5. Acidity raises the rate of corrosion of waters.

Electrolysis and Galvanic Action. Two dissimilar metals in equipment handling water cause galvanic action. This makes the metal of higher electronegative potential in the electrochemical series pass into solution. This should be avoided because chemical treatment cannot counteract this type of corrosion.

Stray electrical currents through piping or equipment may also cause electrolytic action. Stray currents may be caused by high-voltage lines closely paralleling steam or water piping or from improper grounding of electrical equipment. These conditions can be corrected only by eliminating the causes. The corrosion from these sources resembles that caused by oxygen, but few cases of these types occur in power plants.

Steam above about 750 F is corrosive to iron, the chemical reaction producing black magnetic oxide (Fe_3O_4) and hydrogen. This can be avoided by using corrosion-resistant alloys, usually of a high chromium content. These alloys are expensive. In laying out a superheater the designer must make an economic choice between thicker-walled conventional alloys and high-cost corrosion-resistant alloys.

Sometimes corrosion of internal and external surfaces may be avoided by applying *protective coatings.* If the coating should fail, the metal underneath may suffer accelerated local corrosion.

17-7. Internal Boiler Treatments. Small low-pressure boilers using pure make-up water may use *boiler compounds* for an internal chemical water treatment. The compounds may contain caustic soda, soda ash, sodium phosphates or tannin, or starch. These control corrosion and scale. The chemicals react with the hardness to form a precipitate. The organic tannins and starches prevent the precipitates from adhering to the tubes or drums.

Periodic blowdown of the boiler-water circuit removes these precipitates suspended in the water. Each plant has its own specific feedwater problems, best solved by competent analysis chemically; this yields a prescribed treatment for control. Trained chemists are the only ones capable of dealing with these problems.

17-8. Carry-over. Water solids carried over in the steam leaving a boiler drum are called "carry-over." *Foaming* or *priming* of the boiler water forms the carry-over. In foaming, bubbles form on the water surface and may fill the drum steam space. Excessive amounts of sodium alkalinity, or finely divided calcium phosphate, or oil may cause foaming.

Priming refers to the vigorous and periodic surging of water in the boiler drum that throws water slugs into the leaving steam. This may be caused by excessive steaming rate, too high or fluctuating a water level, and improper boiler-water circulation. Carry-over may originate from mechanical or chemical causes.

Carry-over has detrimental results. The water droplets carrying solids evaporate and build deposits in valves, superheaters, and piping and even in the turbine or engine. Superheater deposits reduce heat transfer and raise tube-metal temperatures. Deposits on turbine blading reduce efficiency and capacity and may unbalance the rotor. Deposits forming on governing valves raise a serious hazard.

The purity of steam depends upon its moisture content and the concentration of solids in the liquid. Purity is

$$P = \frac{Y}{100} \times C \tag{17-21}$$

where P = purity of steam, ppm
Y = moisture in steam, %
C = boiler-water solids concentration, ppm

Steam purity is measured from steam samples that are condensed and degassed. The electrical conductivity of condensate as compared with pure water also measures steam purity. For given steam conditions a throttling calorimeter helps solve Eq. (17-21). Recording instruments in large power stations keep constant watch over purity.

17-9. Carry-over Prevention. Carry-over can be corrected only by removing the original cause. For example, carry-over from high water level, sudden load changes, uneven boiler circulation, or basic boiler design can probably be eliminated by preventing or adjusting these factors.

Boiler *drum internals* (Chap. 7) purify steam by removing water droplets and washing with fresh incoming feedwater. The internals are also called "steam purifiers" and "dryers." One type of purifier removes water droplets by centrifugal separation by making the steam flow in a helical path. Another type washes the leaving steam with incoming feedwater. This exchanges droplets of low salt concentration for high-salt-concentration droplets of boiler-drum water. This reduces the solids leaving with the steam.

17-10. Blowdown. As water vaporizes to steam in a boiler, the dissolved solids remain behind to concentrate in the remaining boiler water. All boiler feedwaters, unless distilled, contain some solids. When this concentration in the boiler water exceeds certain limits, the carry-over impairs steam purity. Draining off some of the boiler water carrying excessive concentration of solids and replacing it with fresh feedwater keep the solid concentration within safe limits. This is called *blowing down* the boiler, and the discharged water is called *blowdown*. Required blowdown is calculated by

$$B = \frac{W_B}{W_{FW}} \times 100 \tag{17-22}$$

where B = blowdown, %
W_B = weight of water blown down from boiler, lb per hr
W_{FW} = weight of feedwater supplied for steam supply and blowdown, lb per hr

Modern boiler operation requires that the flow of impurities in the power-plant steam-water circuit be known to ensure safe operation.

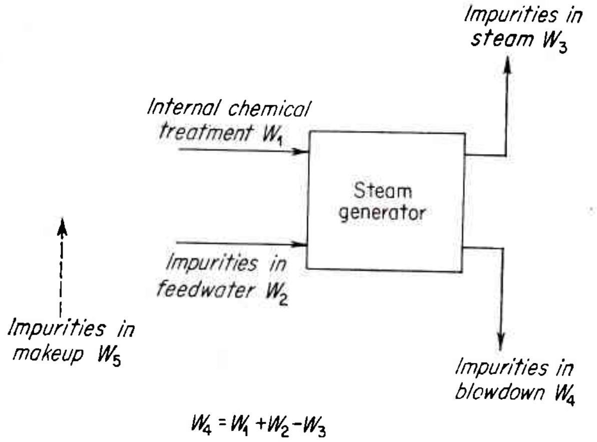
Impurities in steam W_3
Internal chemical treatment W_1
Steam generator
Impurities in feedwater W_2
Impurities in makeup W_5
Impurities in blowdown W_4
$W_4 = W_1 + W_2 - W_3$

FIG. 17-6. Mass balance of impurities and treatment chemicals in steam generator.

Referring to Fig. 17-6,

$$W_4 = W_1 + W_2 - W_3 \tag{17-23}$$

where W_4 = impurities in blowdown = (weight of blowdown) × (allowable salt concentration), ppm
W_3 = steam impurities = (weight of steam) × (impurity concentration), ppm
W_2 = feedwater impurities = (weight of feedwater) × (impurity concentration), ppm
W_1 = internal-treatment chemicals (if used) corresponding to other flow rates

When the main condensate leaving the hotwell is totally pure,

$$W_5 = W_2 \tag{17-24}$$

where W_5 is the impurities in the make-up = (weight of make-up) × (impurity concentration), parts per million. *Make-up* is the difference

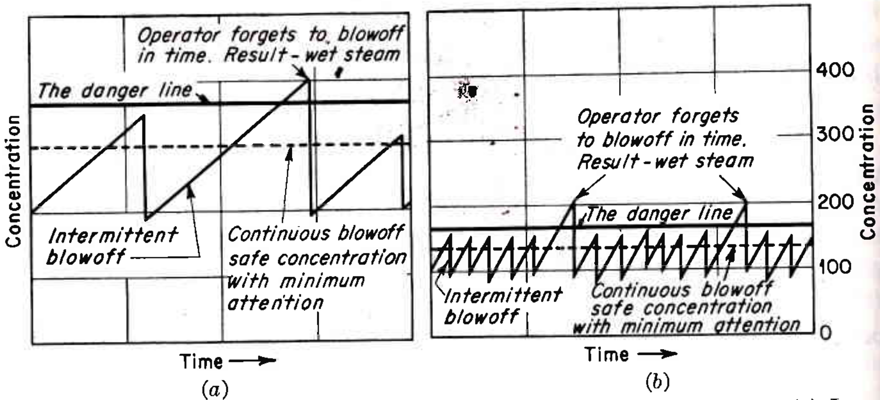

Fig. 17-7. Blowdown patterns for intermittent and continuous systems. (*a*) Low-pressure boilers; (*b*) high-pressure boilers. Note effect if intermittent blowdown is not timed properly. (*Courtesy of The Permutit Company.*)

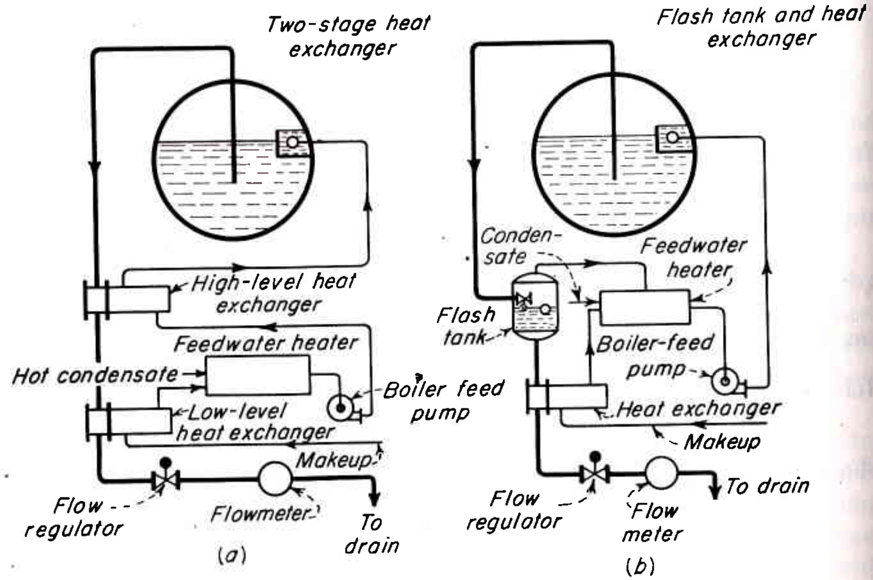

Fig. 17-8. Continuous-blowdown systems are sources of heat recovery. (*a*) Two-level heat recovery reduces irreversibility losses; (*b*) use of flash tank recovers some cycle fluid in addition to heat recovery.

between the amount of water fed to the boiler and the condensate flow from the hotwell. It may be expressed as a percentage of feedwater flow entering the boiler.

Allowable salt concentrations vary mainly with boiler pressure. Table 17-1 shows the limits recommended by the American Boiler Manufacturers Association. Blowdown lines connect to boiler drum, mud drum, or lower headers of furnace waterwalls. Lines from low points remove

Table 17-1. Allowable Boiler-water Concentration Limits

Steam-generator pressure, psig	*Total solids, ppm*
0– 300	3,500
301– 450	3,000
451– 600	2,500
601– 750	2,000
751– 900	1,500
901–1,000	1,250
1,001–1,500	1,000

sludge and dirt accumulations as well as salts. Blowdown may be continuous or intermittent. Continuous blowdown may be some 1 to 10 per cent of the incoming feedwater. Figure 17-7 compares these two systems for both low- and high-pressure boilers. The enthalpy energy of blowdown may be recovered by incoming make-up or condensate in continuous blowdown systems by passing both through heat exchangers. Figure 17-8 shows two methods of recovering blowdown internal energy: (*a*) using high- and low-level heat exchangers and (*b*) using a flash tank and low-level heat exchanger.

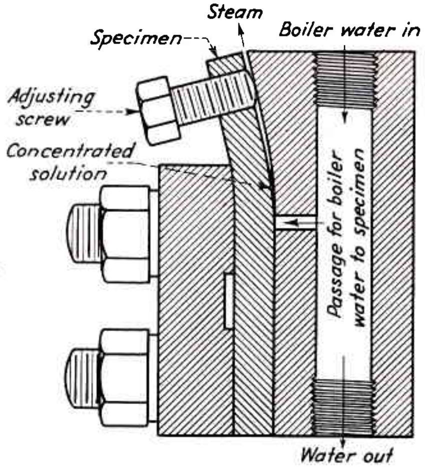

Fig. 17-9. Embrittlement detector as developed by U.S. Bureau of Mines and ASME Joint Research Committee on Boiler Feedwater Studies. Note how the detector reproduces a *stress* condition in the specimen (sample of boiler steel) and a *concentrated* solution of the actual boiler water. These detectors are adjusted daily and are operated over a period up to 90 days to get an averaging effect of the water characteristics.

17-11. Embrittlement. Steels in contact with high-temperature impure waters experience a form of corrosion that disintegrates the grain boundaries of the steel. This appears particularly in riveted seams and stressed tube ends. The intergranular cracks may enlarge to cause structural failure. Welded drum construction eliminates this intercrystalline, or embrittlement, cracking in drums, but tube ends still suffer from this deterioration.

Investigators believe that three conditions contribute to embrittlement: (1) slow boiler-water leaks through cracks or seams vaporize to leave behind a build-up of salts; (2) the metal must be stressed by cold working during fabrication, or riveting, or from expansion and contraction; and (3) the concentrated water in a crack must have chemical properties that will attack the metal.

In plant operation only the water impurities can be varied; inherent stress and minute leakages cannot be controlled. Extensive studies have not yielded a reliable method of predicting the embrittlement characteristics of a boiler water. To measure embrittlement, a detector (Fig. 17-9) is installed in the boiler where boiler water will circulate over it continuously, as in a continuous-blowdown line.

Considerable research has been devoted to embrittlement control. Controls recommended to date include:

1. Elimination of free sodium hydroxide from boiler water
2. Maintenance of a definite ratio of sodium nitrate to sodium hydroxide
3. Using tannins such as quebracho
4. Using waste sulfite liquor
5. Coordinating pH-phosphate control so that the boiler-water pH never exceeds the pH of a trisodium phosphate solution of the same PO_4 content

Application of any of these treatments is followed by continuous checks with the embrittlement detector until a control is established.

PROBLEMS

17-1. A 400-hp boiler operates at 200 per cent rating. Steam condition is 250 psia and 500 F, with feedwater at 250 F. Feedwater contains 8 grains per gal of saline. The boiler-water concentration is to be held to 150 grains per gal. Estimate the hourly blowdown in gallons.

17-2. Calculate the per cent continuous blowdown required for a 1,200-psia boiler which operates with 5 per cent make-up. Impurities in the steam are 1.0 ppm and 70 ppm in the make-up.

17-3. Calculate the relative weights of hydrazine and sodium sulfite required for oxygen removal from boiler water.

17-4. A 450-psia boiler supplies steam of 98.5 per cent quality. Determine the steam purity for standard boiler concentration limits.

17-5. A 650-psia boiler delivers 150,000 lb per hr of steam. A continuous 3 per cent blowdown is used. Steam impurities are 2 ppm with feedwater having 8 ppm. Analyze these data in terms of the type of treatment used.

17-6. To maintain the desired purity of an evaporator output, it is found that 20,500 lb per hr of raw make-up water is required at 70.0 F. The evaporator vapor output of 12,000 lb per hr is delivered to the deaerator heater operating at 126.1 psia. The line drop between evaporator and deaerator is 0.8 psi, and there is a terminal difference in the evaporator of 47 F between shell and tube-side fluids.

a. Compute the required flow of steam from the first extraction stage of a turbine if its original enthalpy is 1351 Btu per lb.

b. What will the flow to waste be, and at what energy level? Could any of this wasted energy be recovered, and if so, how?

c. The evaporator coil drains are likewise delivered to the deaerator heater. At what energy level will they be?

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CHAPTER 18

COOLING-WATER SUPPLY

18-1. Energy Rejection. Even in today's many efficient steam plants at least half the energy input must be rejected as unavailable in the steam-turbine exhaust. The main steam condenser performs the dual function of removing this rejected energy from the plant cycle and keeping the turbine back pressure at the lowest possible level.

The rejected energy inevitably must be returned to the atmosphere. The main condenser does this by transferring the latent heat of the exhaust steam to water exposed to the atmosphere (Chap. 13). This water we call circulating, or cooling, water. Depending on design and plant loading, cooling water warms up by about 5 to 15 F in passing through the condenser; i.e., about 5 to 15 Btu per lb of water is picked up.

Since each pound of exhaust steam gives up about 1000 Btu per lb when condensing and the cooling water picks up only about 10 Btu per lb, we need about 1,000/10 = 100 lb of cooling water per pound of steam condensed. So we need a cooling-water supply about 100 times the flow rate of the steam exhausted to the condenser. For example, a 500-mw steam plant with an exhaust steam rate of 8 lb per kwhr will have a steam flow of 8 × 500,000 = 4,000,000 lb per hr at full load. To carry off the unavailable energy, the plant needs 400,000,000 lb per hr of cooling water. This equals, roughly, 50,000,000 gal per hr—the water needs of a fair-sized city.

18-2. Water Sources. This thirst for cooling water makes water availability a paramount factor in choosing the plant site. Natural sources for large quantities of water immediately suggest the ocean, bays, rivers, and lakes. Plant sites usually stand on or near the shores of one of these bodies of water.

The growing demand for power in many arid areas has led to placing power plants where little water can be found, and that mostly from wells. Such limited supplies make power generation possible by using spray ponds or cooling towers. With these the cooling water follows a closed circuit between condenser and pond or tower. A part of the warmed cooling water evaporates, carrying off the rejected energy to atmosphere. This cools off the bulk of the cooling water that remains in the circuit for return to the condenser.

The water that evaporates at its low partial pressure to atmosphere very nearly equals the quantity of steam condensed in the main condenser. So if a natural water supply from a small river or wells equal to the steam flow in the plant is available, that site will be feasible. Cooling-tower and spray-pond make-up equals exhaust steam flow to the condenser; in addition the plant steam cycle itself will need a 4 to 5 per cent make-up to replace water losses.

18-3. Open-water Circuits. Plants with unlimited water supply have equipment arrangements that vary only in detail from plant to plant, depending on the local problems. Plants on rivers with strong flows have an intake structure, or crib house, on the upstream side of the plant and a discharge, or outfall, on the downstream side.

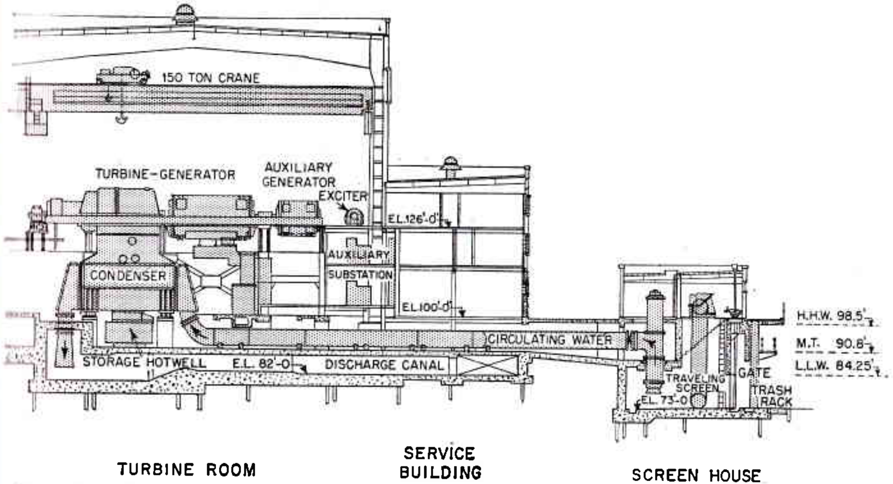

FIG. 18-1. Circulating water enters station from river or bay through screen house at right. Vertical circulating-water pump takes suction in screen-house basin to force water through condenser into discharge canal for return to river or bay by gravity flow.

Plants on lakes and tidal waters have a special problem in avoiding recirculation of the cooling water from the outlet back to the intake. In some plants the intake and outlet are separated as much as 2 miles. This gives ample time for any recirculated water to cool by evaporation before reentering the intake.

Research has shown that warmed cooling water emerging from a plant outfall quickly rises to the surface of the body of water to which it discharges, because of its lower density. It spreads in a thin film on the surface, which aids evaporation and concurrent cooling. A few plants depend entirely on such surface cooling without using sprays or cooling towers. This method proves feasible where enough water surface can be found in natural or artificial lakes and ponds.

Figure 18-1 shows the cooling-water circuit through one tidewater plant. Salt water enters the screen-house well through trash racks that prevent heavy debris such as logs and ice cakes from entering the water circuit. The water then flows under the open unwatering gate; whenever the circulating-water system must be unwatered for repairs, this gate is closed and the water pumped out of the system.

Next the water flows through a traveling screen to remove small debris and fish trapped in the current (Fig. 18-2). These screens usually are controlled automatically; when the water level behind the screen drops a certain amount below that in the front, a pressure-differential gauge energizes the driving motor on top to move the screen up slowly on the front side. This brings clean sections into position from the bottom. As the partly clogged sections move past the upper section, a water spray washes off the debris into a trough that carries it away to waste. The endless chain then carries the cleaned sections down into the water on the far side.

In Fig. 18-1 a vertical slow-speed pump lifts the cooling water from screen-house well and forces it through the circulating water pipe leading to the condenser water box. After passing through the tubing of the condenser to pick up its heat load, the water leaves through the rear water box and the tail pipe to enter the discharge canal. The discharge canal leads the heated water back to the tidal flow at a point several hundred feet from the screen house.

Other layouts vary from this one: some have an intake canal running from the screen house to the base of the condenser with the circulating pump standing next to the condenser; some use horizontal pumps instead of vertical pumps; in many modern plants all the equipment in the screen house stands outdoors. In a multiunit plant there usually is at least one screen for each unit, all standing side by side.

Pumps for house-service cooling for oil coolers, compressor intercoolers, flushing, ash-removal systems, etc., also stand in the screen or crib house. They take their suction after the screen.

Where icing troubles arise in winter, an auxiliary pump or bypass canal may lead some of the exit cooling water back to the inlet. This melts the ice in the intake and prevents clogging.

18-4.* Design Factors—Pumping Cost. The design of a circulating-water system should use minimum cost as a prime criterion; total cost depends on investments and continuing operating costs as explained in Chap. 30. The prime function of a circulating system is to move water, and the costs of this include: (1) investment in pump and driver, which increases as the system head rises; (2) investment in canal, conduit, or

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* Sections 18-4 to 18-10 abstracted from R. T. Richards, Paper 1488, *Journal of the Power Division, ASCE*, December, 1957.

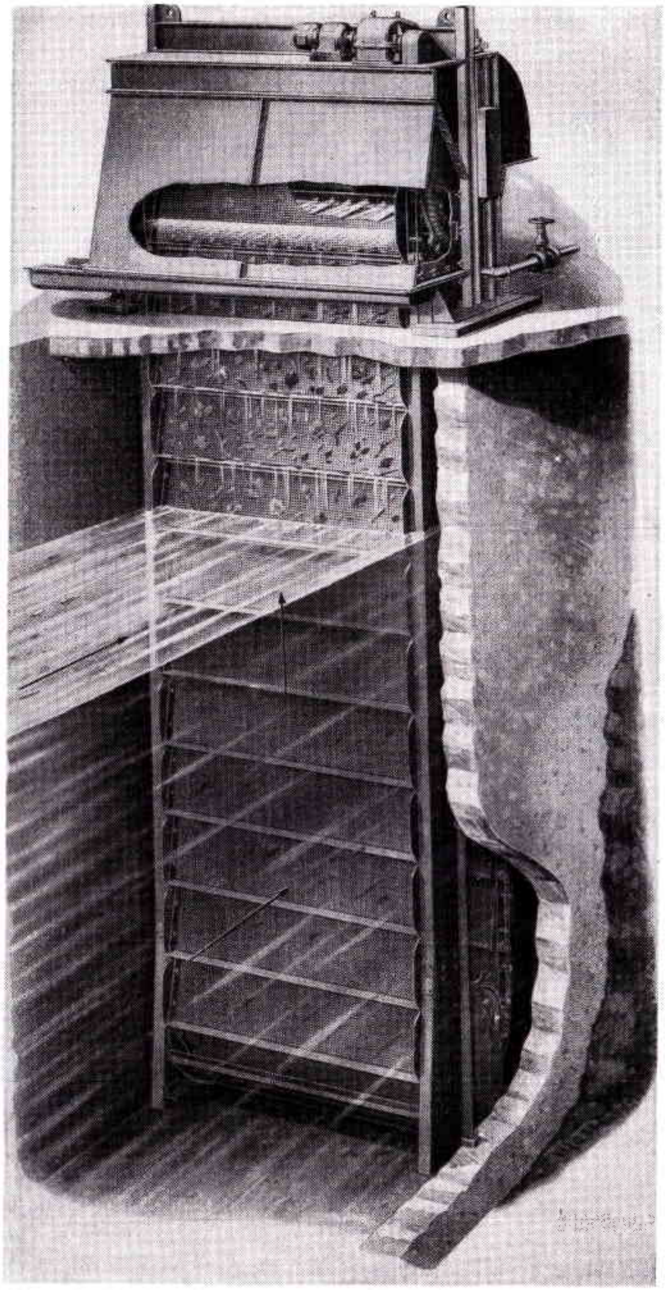

FIG. 18-2. Water-intake traveling screen prevents floating debris from entering station water circuit. Screens in continuous-belt arrangement move up on inlet side as needed to be cleaned by spray at upper end. *(Courtesy of Link-Belt Company.)*

pipeline, which rises with increasing diameters but lowers the pumping-energy cost because of lower friction head; and (3) refinements in water-passage shape and arrangement, which can reduce pumping costs but usually need higher investment.

To arrive at an optimum design, the engineer must know the cost of pumps and drivers, the cost of materials making up the water passages, fixed-charge rates for the plant, and the cost of energy.

18-5. Design Factors—Layout. Arrangement of the circulating-water system largely depends on the location of the water source, screen house, pumps, and main condenser and the subsoil conditions. The layout must be designed to minimize head losses, which concentrate usually at: (1) pump discharges, (2) piping entering and leaving condenser water boxes, and (3) the discharge conduit. These points become critical because the water usually makes sharp changes in direction and speed.

In a good system abrupt transitions, bends, and obstructions must be avoided or minimized. Diffusers at the pump discharge, condenser outlet, and discharge conduit or canal will help to reduce pumping heads. A diffuser consists of a gradual increase in water-passage cross-sectional area that aids in recovering some of the water's velocity head.

At bends a series of curved vanes across the diagonal help turn the water in the new direction with a minimum of turbulence. This reduces over-all friction and resulting pumping cost.

18-6. Design Factors—Pump Head. Intake-water level usually varies over a wide range during a year. This complicates the placing of the pumps. Because of its low frequency of occurrence, pumps are not usually designed for extreme low-water conditions.

Pumps should be placed so that they will run at close to their design point for most of their operating life. Depth of intake structure and length of vertical pumps, however, must be determined from extreme low-water conditions to ensure continuous operation.

Designers usually tend to overdesign a pump to be on the conservative side; this results in a pump that wastes power and may fail prematurely. When a pump runs at higher deliveries, it needs a greater submergence that may not always be available. This may cause cavitation and destructive pitting. Even with ample submergence the pump may run roughly at high flows.

Submergence is the depth of water over the pump impeller; for good design the engineer must know the needed submergence over the entire operating range of the pump. Operators should be aware of the dangers of delivering water to all condensers in a station with some of the pumps out of service.

18-7. Design Factors—Pump Intakes. Horizontal and vertical centrifugal pumps are not particularly sensitive to water-flow patterns at their intakes because the suction pipe usually quiets any turbulence. Vertical axial and mixed-flow pumps, however, are particularly sensitive to vortices in the pump intakes because they may draw air into the pump. This makes the pump run roughly and may pit the condenser tube inlets.

Designers attempt to set up specifications for adequate suction chambers or wells. These vary in results depending on individual station conditions. Because of these erratic results model testing is being used increasingly to find the proper design of hydraulic passages in station circulating-water systems.

18-8. Design Factors—Siphon Recovery. For most plants the discharge level stands below the condenser, and this makes part of the system run with pressures less than atmospheric. For plants at a high elevation above the water source the pumping heads will decrease as the discharge elevations are lowered because of the falling static head. The maximum vacuum that can be held by the system fixes the limit on reduction in pumping head. The maximum vacuum varies with the friction in the discharge conduit. High discharge friction allows higher elevation for condenser settings or lower elevations for system discharges to allow using maximum siphon recovery.

The maximum possible vacuum at sea level equals 34 ft water. But for 97-F water we have a vapor pressure of 2 ft water, and this gives us a net vacuum of 32 ft water which could be used to reduce pumping head. However, two factors prevent realizing this potential: (1) Vacuum pumps to evacuate the system at start-up must be very large to produce this level. (2) Water releases entrained air very rapidly at high vacuums, and this could easily equal vacuum pump capacity. At too high a vacuum need this could cause some of the upper condenser tubes to be water-starved and so could reduce total condenser capacity. Some designers recommend limiting circulating-water vacuum to 28 ft water at sea level and less for higher plant elevations.

The siphon is used to regain pumping head, but it should be kept as low as possible to avoid potential air-binding problems.

18-9. Design Factors—Air Binding. Air may collect in high points of circulating-water systems, and water passages may fail to flow full on downward slopes. This may be corrected by continuously evacuating air accumulated in the system. Unless the air is removed, the pumping head loss may be substantially raised.

Sloping water conduits must be tapped for air evacuation at the top and along the slope at suitable intervals. Mechanical pumps or water- or steam-operated ejectors may be used for evacuation. Air evacuation may have to be continuous or may be needed only at intervals, depending on local conditions.

Air collecting in pipes under pressure usually does not raise serious problems. It can be bled off through taps at the top and a few feet below the top. The taps need be opened manually only during start-up of the unit and only infrequently if at all during operation.

18-10. Design Factors—Water Hammer. Usually, water hammer possibilities grow with circulating-water-system length. Pump start-ups, valve closings, and pump trip-outs usually cause water hammer. Tripping out all pumps together on a system causes the most serious surges.

Following a trip-out, the water leaving the highest point in the system discharges at a rate faster than that supplied by the decelerating pumps. This makes the water column flowing through the system separate and form a vacuum between the two or more sections, which drops the pressure on the leaving column of water. Atmospheric pressure at discharge reverses the flow of the leaving column to fill up the evacuated space.

The colliding columns of water create a severe surge of pressure that strains all parts of the water system. For lines several thousand feet long the pressure surge may reach 300 ft water, many times higher than the design head.

Vacuum breakers, air chambers, open manholes, and other devices may be used to control water hammer. Not many plants have water hammer.

18-11. Water Fouling. Most waters contain living organisms that coat the heat-transfer surfaces and drastically reduce the heat-transfer rates. This seriously reduces the capacity of the condenser and its turbine.

An almost universal treatment for killing algae to prevent slime coating condenser tubes is chlorination of the cooling water. This consists in injecting liquid chlorine or chlorine compounds into the water at intervals frequent enough to kill accumulated algae. Injection points must be carefully selected to ensure adequate mixing.

In some seaboard plants mussels cause difficulty by accumulating on the walls of intake structures to the point of obstructing water flow and increasing the required pumping power. To remove the mussels, the plant must be shut down and the circulating-water system unwatered. This costs money because power must be generated on less efficient stand-by units and labor must be hired to clean the water system.

Mussels can be discouraged by chlorinating the intake water. This must be done during the warm season and must be maintained until the breeding season ends at the onset of cold weather.

Mussels are also killed by water temperatures above 100 F. The Redondo Station of the Southern California Edison Company takes advantage of this weakness. The station, standing on a beach off the Pacific Ocean, takes its circulating water through an intake 1,900 ft

offshore. The water flows through a 10-ft-diameter submerged concrete pipe. Warm-water discharge leaves through a similar pipe and an underwater outlet 200 ft from the intake (Fig. 18-3). To avoid the expense of buying chlorine, the water temperature in the intake line is raised above 100 F at intervals. This is done by manipulating the four gates in Fig. 18-3 that separate the pump-intake basin, transfer basin, and intake and discharge lines. The latter two lines can reverse their function by opening and closing the proper gates. At full station load the discharge water is 19 F hotter than the intake water. To raise the outlet water to 102 F, one of the gates is opened to recirculate just enough water to the condensers. This temperature is held for 4 hr at a time every 10 to 20 days to kill mussels accumulating in the intake line or discharge line.

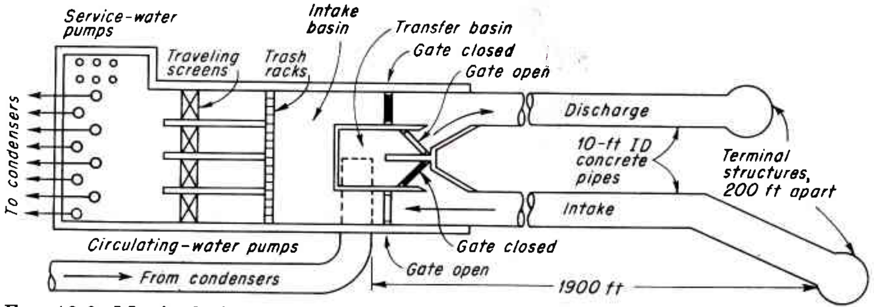

FIG. 18-3. Manipulating four gates between intake and transfer basins allows interchanging 10-ft concrete pipes as intake and discharge to prevent mussel growths; higher discharge-water temperature kills these shellfish when they attempt to establish themselves in pipes.

Some stations on commercial and game-fishing waters are required to avoid killing fish entering their circulating-water system. Fish can sometimes be prevented from entering a water system by stretching an electrified fence across the inlet. The Contra Costa Station of the Pacific Gas and Electric Company uses a fish pump that takes its suction from in front of the traveling screen. It draws in water containing fish up to 14 in. long and discharges them a safe distance from the intake back into the bay. Most fish survive this handling; only 0.2 per cent die.

18-12. Special Designs. Some rivers have high flood stages compared with their normal levels; often they also have high banks. A station on such a river normally would be placed on top of the bank to prevent it being submerged during floods. Occasional plants may be placed on an intermediate level where the site may be flooded at long intervals. These will be protected by dikes that can hold back the highest flood on record.

For a high-bank location the plant must pay a penalty in lifting the circulating water through a high head. To avoid this, some plants have been built with deep condenser pits, about 40 ft below the turbine floor. The turbine discharges to the condenser through a long neck. This causes a slight pressure loss in the exhaust steam, but the saving in cooling-water pumping more than offsets this. The cost of the additional excavation for the deep condenser pits, however, causes an increase in fixed charges of the station.

To avoid deep excavations in situations like these, some plants have reduced pumping costs by recovering some of the pumping energy by making the outlet water do work in a hydraulic turbine. The turbine helps drive the pump rotor and so reduces the energy needed from the driving motor.

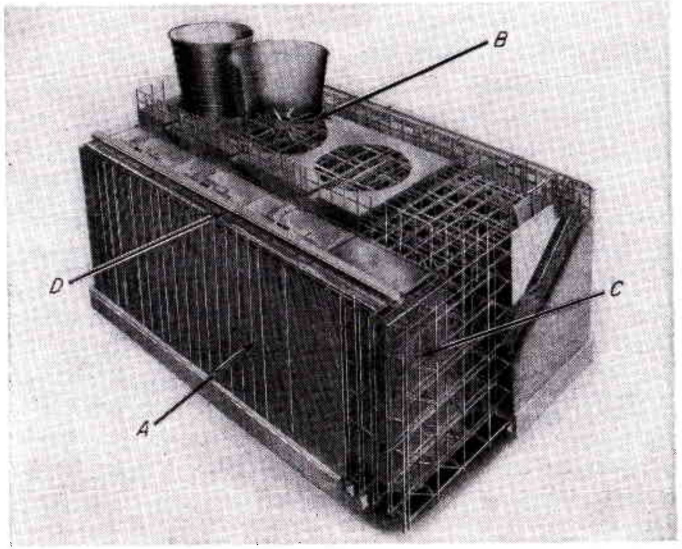

Fig. 18-4. Induced-draft (i-d) cooling tower receives water through distributing header *D* which flows through holes in pans and falls over wood filling *C* to collecting basin in tower bottom. Air enters through louvers *A* to pass over wood filling and discharges to atmosphere through i-d fan *B*. (*Courtesy of The Marley Company.*)

18-13. Cooling Towers. More power stations are being built in water-short areas. Lack of large water flows makes these plants rely on cooling towers to remove heat energy from the circulating water flowing in a closed circuit.

Of the variety of cooling-tower designs available the induced-draft (i-d) tower finds the widest application in central stations. Figure 18-4 shows sections through a four-cell tower. Air enters the side louvers to flow around the wood filling and up through the i-d fans at the top of the tower.

Hot circulating water from the condenser outlet enters the distributing header at the tower top to flow into the distributing basins forming the roof of the tower. From these the water flows through many holes to fall on the wood-fill strips below. This breaks up the water into small drops and spreads the water in thin films on the fill surface as it cascades to the collecting basin at the bottom.

As the water falls in drops from strip to strip, it exposes a maximum surface to speed evaporation and heat transfer. The drops fall through the cross- and counter-air flow that carries away the vapor and heat as they are released from the water surface.

The water cools by (1) evaporation and (2) heat transfer to the air. In many towers about 75 per cent of the cooling takes place by evaporation and the remainder by heat conduction; the ratio depends on the humidity of the entering air and various design factors.

Vapor separating from a nonboiling fluid always leaves behind a cooler liquid because the vapor is composed of the higher-energy molecules originally in the liquid. In addition the cooler air picks up heat by conduction from the warmer water.

Tower cooling performance is always referred to the wet-bulb temperature of the incoming air. This is the lowest temperature that the outgoing water can be cooled to. The finite dimensions of a tower and the limited time in which water and air contact each other make it impossible to achieve this ideal cooling.

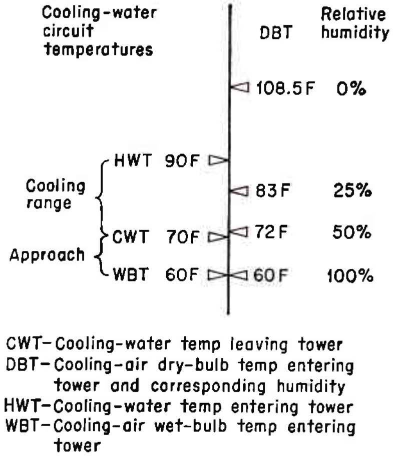
Cooling-water circuit temperatures
DBT
Relative humidity
108.5F 0%
HWT 90F
Cooling range
83F 25%
CWT 70F
72F 50%
Approach
WBT 60F
60F 100%
CWT-Cooling-water temp leaving tower
DBT-Cooling-air dry-bulb temp entering tower and corresponding humidity
HWT-Cooling-water temp entering tower
WBT-Cooling-air wet-bulb temp entering tower

Fig. 18-5. Difference between exit-cooling-water temperature and wet-bulb temperature of entering air is the approach; the smaller the approach, the more efficient the tower. Diagram of a typical tower shows various dry-bulb temperatures and corresponding humidities that equal 60-F wet-bulb air.

The principal performance factor of a tower is its *approach* to the wet-bulb temperature; this is the difference between the cold-water temperature leaving the tower and the wet-bulb temperature of the entering air. The smaller the approach, the more efficient the tower. Another important performance factor is the *cooling range;* this is the difference between the hot-water temperature entering the tower and the cold-water temperature leaving. Figure 18-5 gives a graphic demonstration of these factors for a given tower.

The scale to the right of Fig. 18-5 shows the dry-bulb temperatures and relative humidities that correspond to a wet-bulb temperature of

60 F. If the humidity is low enough, a tower can cool water very effectively even with a high dry-bulb temperature.

Make-up must be continuously added to the tower collecting basin to replace the water lost by evaporation and spray carry-over. As mentioned before, this nearly equals the amount of steam exhausted by the turbine. The circulating-water circuit must be blown down periodically to remove the accumulated solids concentrated by the evaporation. The water must be treated to kill algae, preserve the wood fill, and prevent metal corrosion.

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CHAPTER 19

STATION CONTROLS

19-1. Load Variation. Most power-plant loads vary in an irregular manner; the operators can usually anticipate the hour-to-hour changes but not their exact values. The fuel and air feed to boiler furnaces must be varied accordingly, or steam pressures and turbine shaft speeds will get out of hand.

Figure 19-1 shows what happens when load changes on a station. Before time *A* the plant carries a steady load, and energy input exactly balances plant output and losses. At time *A* a bank of lights is switched on the system.

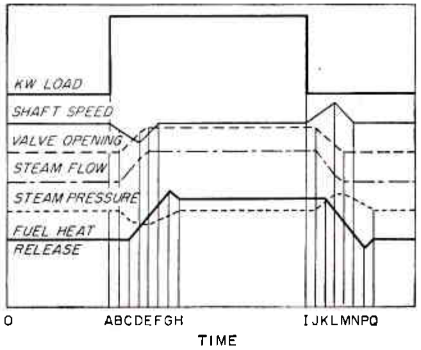


Fig. 19-1. Changes in electric load upset the energy-flow balance in a central-station cycle and cause changes in shaft speed, valve openings, and steam pressures.

This reduces the over-all electric resistance of the system, and a greater current flows through the generator armature and connected circuit. Higher amperage raises the armature magnetic-field strength, increasing the turning resistance offered the field rotor; this slows the generator and turbine shaft speed. By giving up some of its rotating kinetic energy the shaft momentarily supports the added load.

The drop in shaft speed at time *B* brings the turbine governor into action, and the turbine steam-admission valves begin to open wider. More steam immediately flows into the turbine. Because energy input to the boiler furnace has not changed, steam pressure begins to drop. The pressure drop allows some of the boiler-drum water to flash to steam, which momentarily helps to carry the added load by drawing on internal energy stored in the boiler water.

The pressure reduction signals an automatic combustion-control system to speed up the forced-draft (f-d) and induced-draft (i-d) fans and the fuel-feed system at time *C*. The increased heat release in the furnace steps up the rate of steam generation and checks the rate of pressure fall. At time D the governor reaches a balance point to hold the higher load at a speed slightly less than normal. Adjusting the governor-spring tension opens the throttle valve a little wider at time E, further increasing steam flow and bringing shaft speed back to normal at time F. As pressure nears its rated value at G, the firing rate eases off, overshooting slightly to restore balanced conditions more quickly at time H. The total time interval from A to H depends on the size of the load change and may take only seconds or a few minutes.

Switching off the bank of lights at I increases the system resistance and reduces the amperage. This reduces the opposing force set up by the diminished armature magnetic field and allows the turbine-generator rotor to speed up. The overspeeding rotor absorbs the excess energy input to the turbine and signals the governor to reduce the opening of the steam-admission valve at time J. This reduces the steam flow, which raises the boiler steam pressure. Rising steam pressure signals the automatic combustion-control system to slow down the f-d and i-d fans and fuel feed into the boiler furnace at time K.

At L the governor reaches a new balance point where it lets the turbine carry the lower load at a slightly high shaft speed. Adjusting governor-spring tension closes the throttle valve further, lowers steam flow slightly, and brings shaft speed back to normal. The decreasing firing rate has by this time allowed pressure to rise through a peak and drop toward its normal value. As it approaches this level, the firing rate eases upward to keep pressure and other factors steady after time Q.

Other adjustments must be made in the plant cycle during load changes, for example, maintaining water levels in boiler drum, condenser hotwell, and feedwater heaters, holding furnace pressure, and changing pump speeds, as well as pulverizer and oil-pump speeds. There are so many of these that it becomes impossible for a crew of operators to coordinate them manually. Automatic controllers perform these functions so that many modern units need only two or three operators per shift.

19-2. Control Arrangements. These vary considerably from plant to plant. In older plants instruments and controls were generally grouped on a panel next to the equipment. Operators had to move from unit to unit to start and stop them and check their operating condition. From this scattered arrangement, which needed as many as 8 to 10 operators for proper supervision, the modern station has evolved a grouped arrangement of instruments and controls. In the *area system* all the controls and instruments for the boiler and its auxiliaries are brought to one room or area in the boiler room, all the controls and instruments for the turbine and its auxiliaries are brought to one room or area in the turbine room, and all the controls and instruments for the generator and its auxiliaries are brought to one room in the switch house or yard.

In the *centralized control system* the controls and instruments for all station equipment are placed in one room under the supervision of one to three operators. Both arrangements use automatic controls on practically all equipment to minimize the need of manual supervision. Figure 19-2 shows a centralized control room for one unit in a modern central station. Below the ceiling are annunciator panels with squares that light up when equipment, pressures, temperatures, speeds, etc., approach abnormal conditions. A bell or howler calls the attention of the operator.

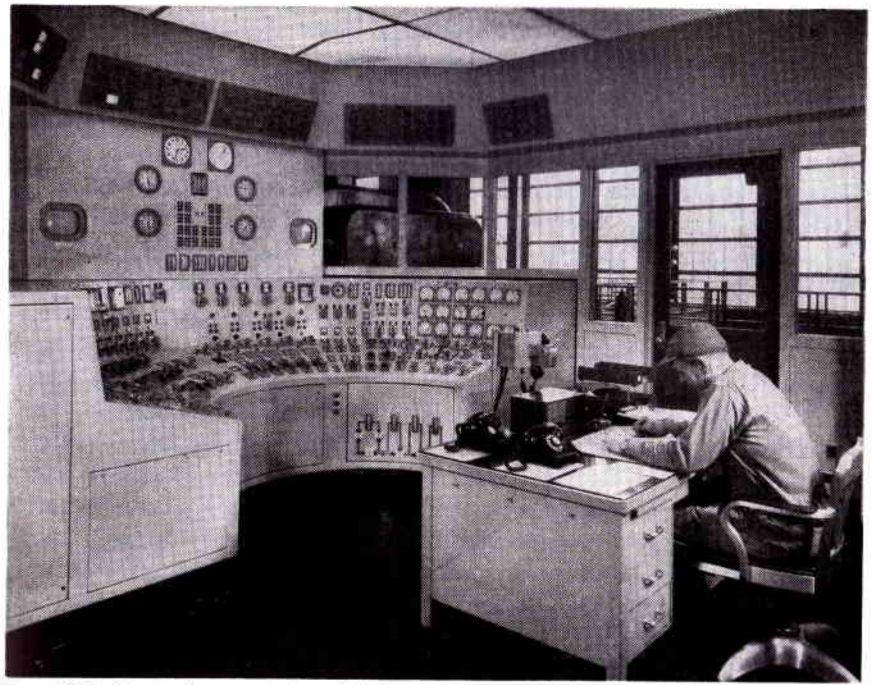


FIG. 19-2. Central control board in a modern central station uses annunciators, miniature instruments, and control console for the entire station. One operator can control the entire unit, boiler, turbine, and generator switching. (*Courtesy of Public Service Electric and Gas Company.*)

On the vertical panel are a clock, total kilowatt load indicator, pressure gauges, indicating thermometers, buttons and lights for controlling the boiler soot-blower system, and two television screens for observing furnace conditions and boiler-water level. The console carries recording and indicating meters for flows, pressures, temperatures, and currents for the boiler and turbine and their auxiliaries. The various handles and buttons give the operator remote control of practically all station equipment, including dampers and valves. Pilot lights tell the operator the position of the controlled equipment. On the far right of the console are the ammeters, voltmeters, power-factor meter, wattmeter, and control handles for the main generator and its exciter.

19-3. Combustion Control. Automatic combustion-control systems are available in a number of arrangements, but they all essentially follow a common mode of operation. They must have a detecting element sensitive to a change in steam demand. To maintain best steam-generator efficiency, the fuel-air ratio must be regulated while load changes.

Most control systems measure steam pressure to detect changes in steam demand. Then, through a relaying system, signals generated by changing steam pressure cause control drives to adjust motor-drive speeds and damper and valve positions to satisfy the new demand. Measurements of air and fuel flows then make secondary adjustments to the control system to keep optimum fuel-air-ratio feed to the boiler.

Here are some of the arrangements used by control systems:

1. Measure steam pressure to control air flow: measure air flow to control fuel flow.
2. Measure steam pressure to control fuel flow: measure fuel flow to control air flow.
3. Measure steam pressure to control both air and fuel flows.
4. Measure steam pressure to control both air and fuel flows: measure both steam flow and air flow to make fuel-air-ratio adjustment on air flow.
5. Measure steam pressure to control both air and fuel flows; measure both steam and air flows to adjust fuel-air ratio by varying fuel flow.
6. Measure steam pressure to control fuel flow: measure steam flow and air flow to control air flow.
7. Measure steam pressure to control air flow: measure steam and air flows to control fuel flow.
8. Measure steam pressure to control fuel flow: measure steam flow to control air flow.
9. Measure steam pressure to control fuel flow: measure smoke density to control air flow.
10. Measure governor speed-changer position to control fuel flow: measure steam flow to control air flow.

For oil and coal, flow measurements prove difficult to make accurately. Basically we need to measure furnace heat release. Experience shows that steam flow proves to be a good index of fuel flow, even with small changes in boiler efficiency with load.

To minimize fluctuations in system frequency, the best scheme would theoretically start changing heat release in the furnace before the load change, because of the lag in steam-generator processes. Scheme 10 does this to a degree by measuring governor action; this is used as an anticipating element.

19-4. Combustion-control Equipment. The basic parts of a control system include: (1) primary elements that respond to cycle changes, for example, steam pressure; (2) relaying elements that convert these responses to a controlling impulse and transmits it to the control points; and (3) power units that receive the control impulse and provide "muscle" to move dampers, adjust rheostats, close and open valves, etc.

In most systems steam pressure actuates the primary element of the master controller. The relay system may be: (1) a simple mechanical arrangement directly controlling motive fluid for the power units, (2) a piping layout using variable air pressures, (3) an electrical circuit using solenoids or contacts, or (4) an electronic circuit with amplification of small electrical currents.

19-5. Control Requirements. Controllers must meet several criteria: (1) sensitivity, (2) speed, (3) stability, (4) power, and (5) ruggedness. Maintaining reasonably constant steam pressure requires enough sensitivity to detect small steam-pressure changes. This allows initiating corrective movements of fuel and air controls soon enough. Both air and fuel flows need time to speed up or slow down to meet the new demand level for steam.

The total time lag of a control system has three parts: (1) time for the master regulator or controller to respond to steam-pressure change, (2) time for the fuel feeder and fan speed or damper to reach the new value or setting, and (3) time for the steam-pressure change to respond to the new input rate. A speedy response of the control system greatly helps to reduce the first two time intervals.

A stable control system changes the input smoothly to meet the new load without letting steam pressure fluctuate about the desired level. An unstable system keeps hunting, even when the new load conditions have been established, because of too much time lag. A good system strikes a happy medium between response speed and stability.

Available systems meet these conditions in various ways. Compensating devices using dashpots, springs, resetting arrangements, floating pivots, etc., cause the master controller to approach a temporary balance before steam pressure returns to normal. This limits the amount of resetting of fuel and air controls that the initial steam-pressure deviation calls for, preventing them from going to extreme positions, or overtraveling. As steam pressure returns to normal, the small residual unbalance of the controllers makes small corrective movements that bring the boiler to its new rating smoothly.

The power units must have ample force to move fuel and air controls as demanded. Power must be available to overcome accidental friction or sticking.

Though regulators must be sensitive to small forces, they also must be

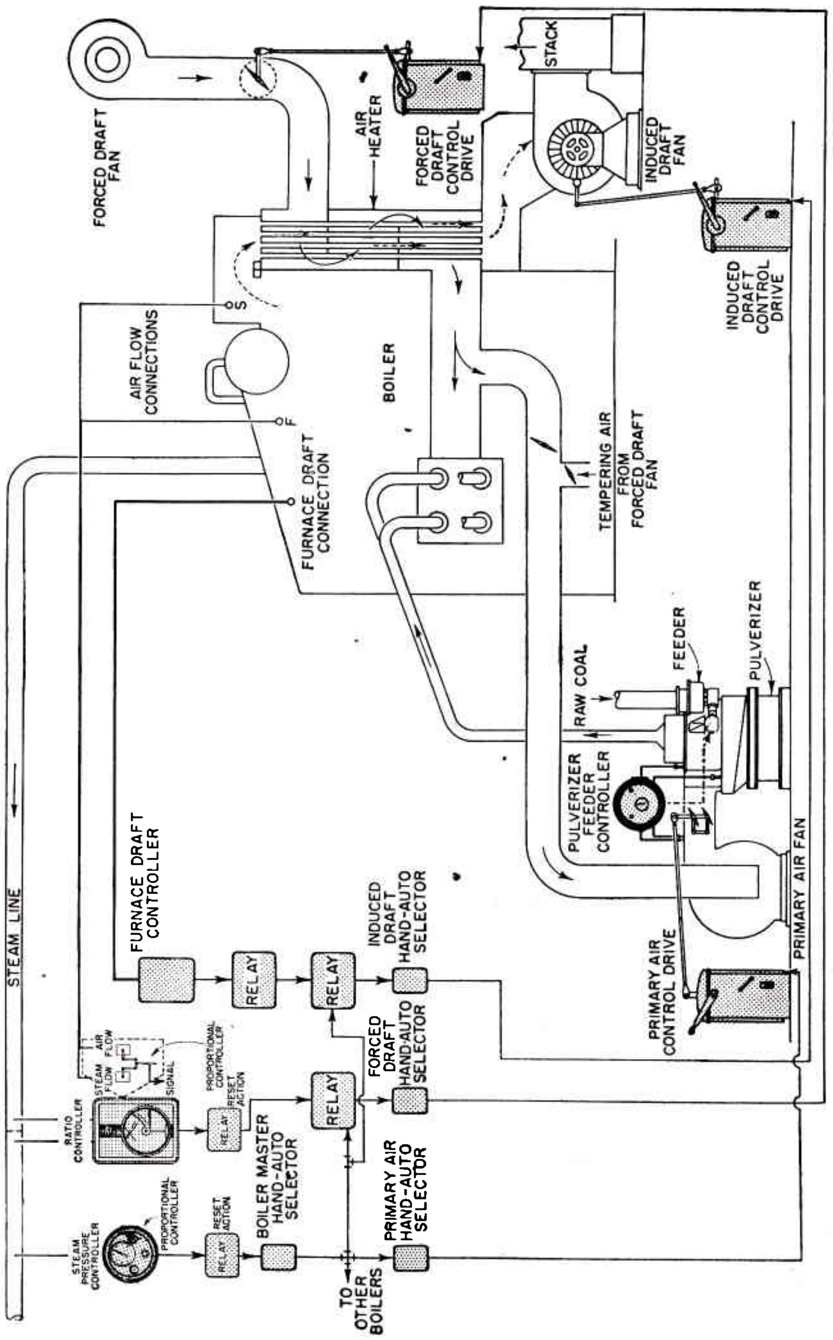


FIG. 19-3. Pulverized-coal-fired steam-generator control system uses pressurized air as relaying medium. System regulates coal flow through control of primary air at pulverizer; regulates air flow at forced-draft- (f-d) fan outlet jointly with steam-flow air-flow meter, which holds fuel-air ratio. Furnace pressure controls inlet-air damper position or induced-draft- (i-d) fan speed. (*Courtesy of Bailey Meter Company.*)

rugged enough to withstand continuous operation in an environment that is often difficult.

19-6. Compressed-air Control System. Figure 19-3 shows a combustion-control system that uses compressed air for its relaying system and its power units. The steam-pressure controller measures the steam pressure and adjusts the relaying air pressure accordingly through the relay valves below. The ratio controller measures fuel flow in terms of steam flow from the boiler and the air flow through the furnace in terms of the pressure drop of gases flowing through the furnace. The floating linkages, shown schematically, combine these two measurements and adjust the connected relays to send a modified relay air pressure to move the power unit of the f-d-fan control drive. This opens or closes the damper as needed by the load.

The furnace-draft controller, through the compressed-air relay system adjusts the i-d control drive to maintain the furnace draft at a preset level. This adjusts the fan-inlet-vane positions or regulates the fan-drive speed.

The relayed air pressure from the steam-pressure controller also acts on the primary air-control drive for the coal pulverizer. The change in primary air flow makes the pulverizer feeder controller vary the feeder speed with the changing steam flow.

The selector valves allow an operator to remove the controls from supervision of the automatic system and remotely control these auxiliaries from the panel board in the control room. Units usually start and stop under manual control but carry normal loads under automatic control.

19-7. Electric-type Control System. Figure 19-4 shows a combustion-control system that uses change in generator electric load, as measured by the recorder, as the first impulse to alter steam-generator fuel and air flow. The steam-pressure "master controller" acts to adjust fuel feed as needed, in parallel with the load recorder impulses.

Through the mixer amplifier anticipatory signals are sent to the governor motor in parallel with those sent to the fuel-feed controller. These impulses make the boiler fuel input change simultaneously with changes in governor motor position even before the generator changes are fully made. As generator output responds to the new load demand, the fuel input keeps in step. To maintain proper fuel-air ratio with changing load, the O_2 recorder sends adjusting impulses to the air-flow controller. The O_2 setter allows adjusting the control system for the best combustion-gas conditions with changing fuels.

The furnace pressure controller acts in parallel with the air-flow controller to adjust the i-d-fan inlet and outlet dampers.

This system has four elements: (1) governor motor impulses—anticipatory action; (2) load (generator output)—proportional action; (3) steam

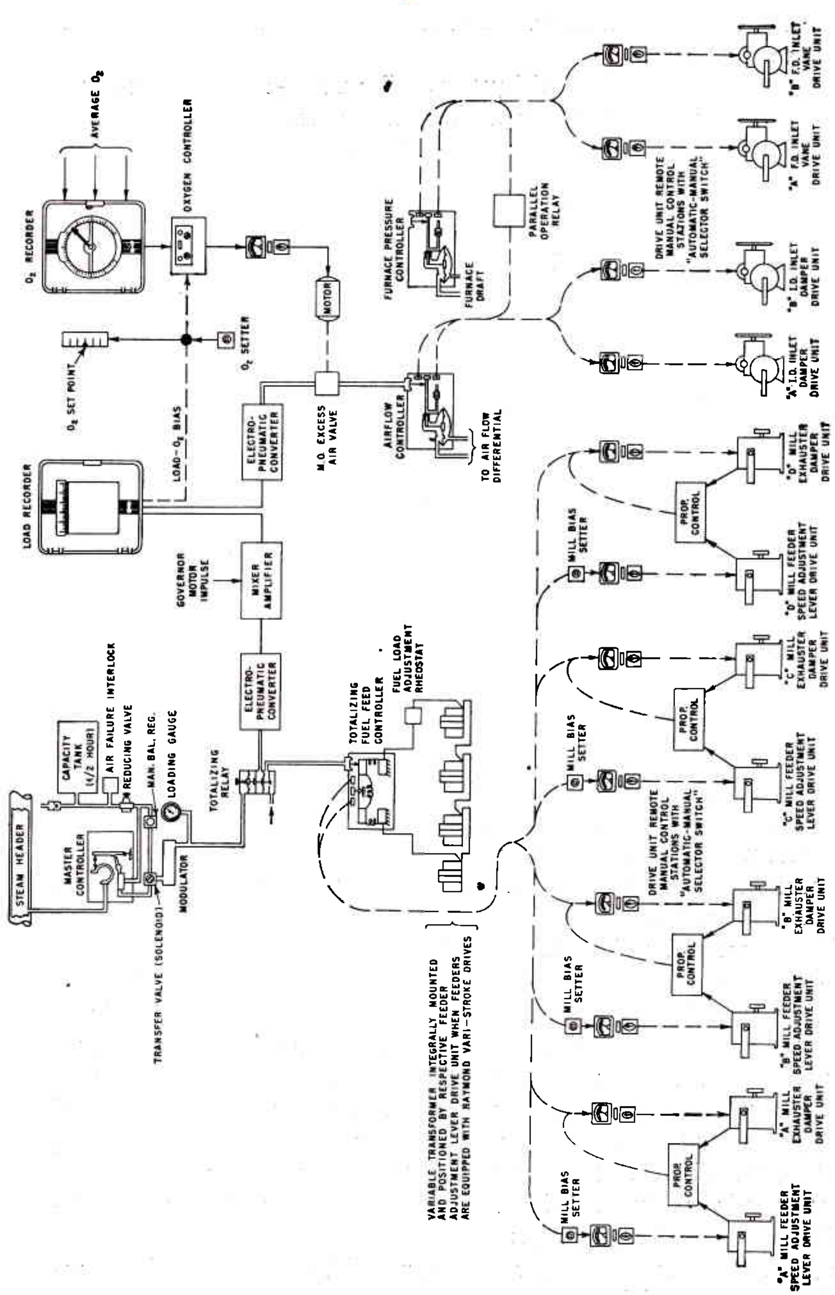

O₂ RECORDER
AVERAGE O₂
OXYGEN CONTROLLER
O₂ SET POINT
O₂ SETTER
LOAD RECORDER
LOAD-O₂ BIAS
MOTOR
ELECTRO-PNEUMATIC CONVERTER
M.O. EXCESS AIR VALVE
GOVERNOR MOTOR IMPULSE
MIXER AMPLIFIER
ELECTRO-PNEUMATIC CONVERTER
STEAM HEADER
MASTER CONTROLLER
CAPACITY TANK (1/2 HOUR)
AIR FAILURE INTERLOCK
REDUCING VALVE
MAN. BAL. REG.
LOADING GAUGE
TOTALIZING RELAY
MODULATOR
TRANSFER VALVE (SOLENOID)
TOTALIZING FUEL FEED CONTROLLER
FUEL LOAD ADJUSTMENT RHEOSTAT
FURNACE PRESSURE CONTROLLER
FURNACE DRAFT
PARALLEL OPERATION RELAY
AIRFLOW CONTROLLER
TO AIR FLOW DIFFERENTIAL
DRIVE UNIT REMOTE MANUAL CONTROL STATIONS WITH "AUTOMATIC-MANUAL SELECTOR SWITCH"
"B" F.D. INLET VANE DRIVE UNIT
"A" F.D. INLET VANE DRIVE UNIT
"B" I.D. INLET DAMPER DRIVE UNIT
"A" I.D. INLET DAMPER DRIVE UNIT
"D" MILL EXHAUSTER DAMPER DRIVE UNIT
PROP CONTROL
MILL BIAS SETTER
"D" MILL FEEDER SPEED ADJUSTMENT LEVER DRIVE UNIT
"C" MILL EXHAUSTER DAMPER DRIVE UNIT
"C" MILL FEEDER SPEED ADJUSTMENT LEVER DRIVE UNIT
"B" MILL EXHAUSTER DAMPER DRIVE UNIT
"B" MILL FEEDER SPEED ADJUSTMENT LEVER DRIVE UNIT
"A" MILL EXHAUSTER DAMPER DRIVE UNIT
"A" MILL FEEDER SPEED ADJUSTMENT LEVER DRIVE UNIT
VARIABLE TRANSFORMER INTEGRALLY MOUNTED AND POSITIONED BY RESPECTIVE FEEDER ADJUSTMENT LEVER DRIVE UNIT WHEN FEEDERS ARE EQUIPPED WITH RAYMOND VARI-STROKE DRIVES

FIG. 19-4. Complete metering-type combustion-control system uses combination of pressurized air and electrical circuits to control fuel and air feed to boiler. System receives control impulses from generator load recorder, steam-pressure master controller, and O₂ recorder. (*Courtesy of Leeds & Northrup Company.*)

pressure—final adjustment, or "trim"; and (4) automatic correction of excess air from flue-gas O_2. The system uses electric-drive units that self-lock on power failure. They receive electric impulses from the controllers to direct their actions in adjusting motor speeds or damper settings.

The figure does not show the selector switches and valves that allow operators to run the auxiliaries manually by remote control or at the equipment.

19-8. Feedwater Control. Feedwater flow must be varied with change in station load. Several systems are available for automatically controlling feedwater flow: (1) single-element self-operated, (2) single-element pilot-operated, (3) two-element pilot-operated, and (4) three-element pilot-operated.

Figure 19-5 shows a single-element pilot-operated feedwater-control system which allows manual correction of water level with flow variation. The water-level recorder operates an air pilot valve, which through a relay and indicator controls the feedwater-regulating valve. The automatic reset of the pilot valve makes it possible to hold drum level constant at all rates of steam flow. Manual adjustment at the selector valve allows raising the water level at high load, and vice versa.

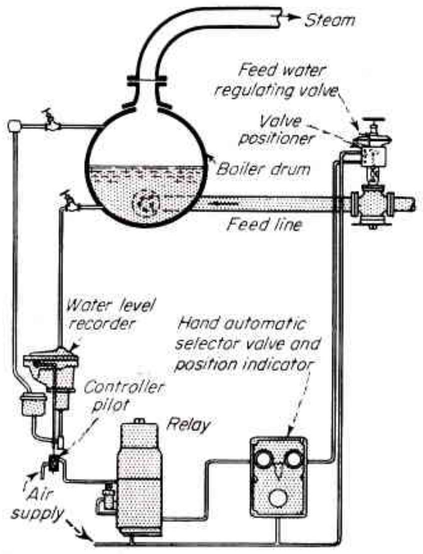


FIG. 19-5. Single-element pilot-operated feedwater regulator holds level constant in boiler drum. Level may be adjusted manually. (*Courtesy of Bailery Meter Company.*)

Figure 19-6 shows a three-element pilot-operated system using an additional feedwater-flow measurement. The ratio of steam and water flows exerts primary control, with readjustment by the level measurement. Changes in water flow from change in water pressure can be compensated by adjusting the feedwater valve. Practically all large boilers use this control system.

19-9. Steam-temperature Control. This plays an important part in modern central stations. Control methods have been outlined in Secs. 8-1 and 8-2. Time lags in temperature measurement make automatic control systems difficult to design, and yet temperatures must not be allowed to exceed maximum limits or they may cause severe damage. Thermocouples of the high-speed type take at least 48 sec to respond to 90 per cent of a given change in temperature in their indication—far from instantaneous.

Boiler air flow is the simplest and most common element used for anticipating action in steam-temperature control. Except for boilers with large radiant superheaters steam temperatures normally increase with higher combustion rates. Boiler air flow directly indicates combustion rate and should assist control in the proper direction. Air flow can also be measured with minimum time delay.

Most reheat units have at least two methods of controlling one of the temperatures. This requires sequence control of two actuators from one controller. Lag characteristics may be entirely different for the two methods of control. Adjustments must be in the equipment as controller transfers from one actuator to another.

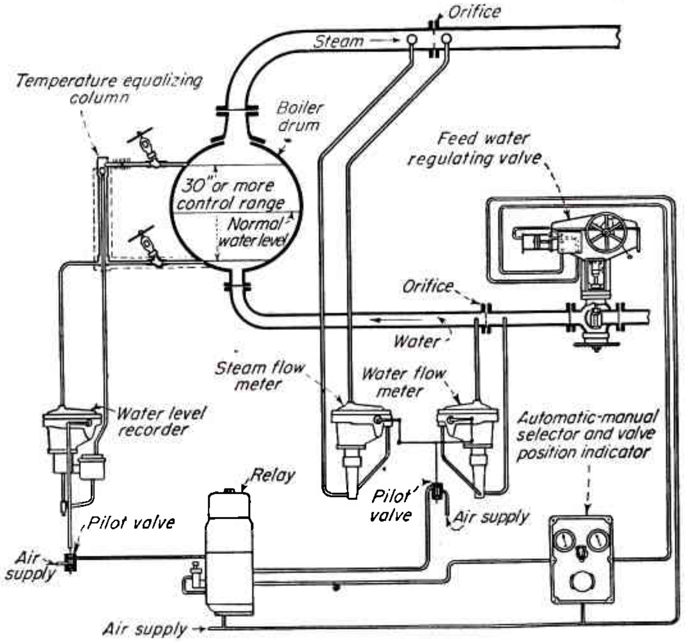


FIG. 19-6. Three-element pilot-operated feedwater control keeps the water and steam flow balanced and adjusts the water level as needed to keep it within safe range. (*Courtesy of Bailery Meter Company.*)

On some reheat steam generators regulation of each of the temperatures affects the other temperature. For example, the amount of hot gas recirculated affects both reheat and superheat temperatures, and so will any movement of the proportioning dampers within the boiler. Because of this some automatic temperature controls cost more than automatic combustion-control systems.

19-10. Controls and Instruments. Controls and instruments located in modern central-station control rooms include controls for the following equipment: station-service air compressor, instruments-and-controls air compressor, soot-blowing air compressor, fire pumps, intake screens, circulating-water chlorination system, feedwater-treatment system, step-up transformer breaker, auxiliary-power-supply breakers, transmission-line breakers, automatic load control, automatic frequency control, and automatic voltage regulator.

Station control boards usually have instruments for the following:

Auxiliary bearing temperature
Drum level
Steam temperature
Furnace flame condition
Furnace outlet O_2
Furnace outlet CO_2
Feedwater pressure
Steam-drum pressure
Superheater outlet pressure
Air-heater inlet-air temperature
Air-heater outlet-air temperature
Boiler-drum steam temperatures
Coal-air-mixture temperature
Exciter air temperatures
Economizer inlet-gas temperature
Economizer outlet-gas temperature
Air-heater gas temperatures
Superheater gas temperatures
Reheater gas temperatures
Economizer water temperatures
Main steam temperature
Reheat steam temperatures
Superheater steam temperatures
Reheater steam temperatures

Boiler equipment controlled from the central room also includes: soot blowers, coal-handling system, dust- and ash-handling systems, i-d and f-d fans, boiler feed pumps, pulverizer, coal feeders, burner ignition system, secondary-air dampers, and chemical feeders. Boiler controls may include interlocks to trip the boiler out of service for: loss of f-d or i-d fans, loss of turbine, loss of fuel, high positive or negative draft, and low boiler circulating-water pressure drops.

The turbine auxiliaries controlled include: condenser circulating pumps, hotwell pumps, bearing oil pumps, make-up pumps, evaporator, gland water flow, gland steam flow, and turning gear.

Instruments also make measurement of the pressure and temperature of: the drum steam, attemperator, reheater inlet and outlet, turbine first-stage steam, extraction steams, turbine exhaust steam, circulating water in and out of the condenser, atmosphere, hotwell condensate, condensate pumps, heater condensates, boiler feed-pump suction and discharge, heater feedwater, drips from each heater, economizer inlet and outlet, evaporator gland steam, and deaerator. Turbine supervisory instruments include: the shaft-eccentricity recorder, speed and camshaft recorder, vibration-amplitude recorder, shell and differential expansion recorder, spindle position, and bearing-pedestal vibration.

Main turbines may be interlocked to trip off the line for: overspeed, loss of vacuum, abnormal thrust-bearing condition, low lube oil pressure, high exhaust-hood temperature, loss of field, electrical grounds, low hydraulic oil pressure, loss of load, loss of i-d fans, opening of generator breaker, closing of stop and intercept valves, excess vibration, loss of fuel, generator overcurrent, and main-transformer high temperature. Some stations provide for temperature monitoring of motor windings, motor bearings, and pump bearings.

19-11. Automatic Tripping. Figure 19-7 shows the tripping scheme developed for the Astoria Station of the Consolidated Edison Company of New York to remove equipment automatically from service in the event of trouble. The turbine generators are cross-compound, each fed by a single steam generator.

Thrust-bearing Failure. Thermocouples inserted in the metal backing of the babbitted thrust shoes give fast response to changes in metal temperatures. The thermocouples actuate an alarm at 10 F above normal; at 190 F they operate the auxiliary relay to trip out fuel supply and the turbine stop valve.

Excessive Vibration. Vibration velocity pickups on the turbine-bearing pedestals actuate the auxiliary relays to trip turbine and boiler if the turbine rotors should become unbalanced from heavy rubs or throwing a blade.

Vibration pickup on the outboard-bearing pedestal of the generator protects it from mechanical unbalance, such as failure of the cooling fan, and also from electrical unbalance caused by double-winding grounds in the rotor. These cause an unbalanced magnetic pull on two sides of the rotor. Short-circuit currents in the rotor may cause unequal heating of the rotor and consequent mechanical unbalance by bowing the shaft.

Exhaust-hood and Condenser-shell Heating. Sprayed water cools the exhaust steam at low loads to prevent overheating the exhaust hood and condenser shell. Two thermocouples in the exhaust casing measure the metal temperatures, which depends on the steam temperature. One thermocouple connects to a recorder and the second to a trip controller. The recorder has contacts for an alarm circuit, and the controller has contacts for the trip circuit.

Loss of Vacuum. Rupture diaphragms in the exhaust shell prevent positive pressure in the condenser on loss of circulating water. To trip the machine in advance of disk rupture, a vacuum-measuring device is used. This sounds an alarm at 27 in. Hg vacuum and trips out the boiler and turbine at 20 in. Hg vacuum.

Shaft Overspeed. A conventional overspeed governor on the turbine shaft trips the main turbine stop valve, governor valves, and reheat intercept valves.

Quick Starting. These reheat units have quick-starting bypass valves around the main and reheat cylinders to the condenser to establish mass steam flow through superheater and reheater so that both main and

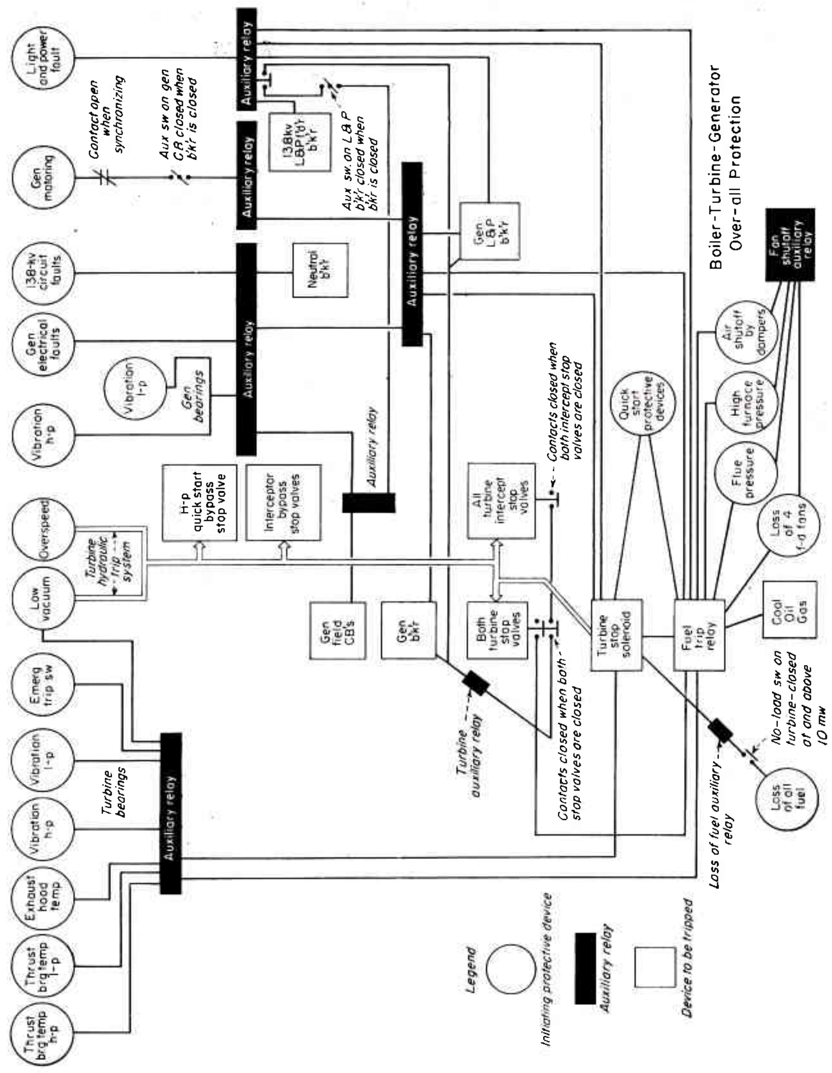


FIG. 19-7. Protection scheme for modern, large central-station unit shows comprehensive coverage against accidental breakdowns or misoperations.

reheat steam temperatures are about 100 F above the chest-metal temperature of the respective cylinders before opening the turbine control valves. Automatic trip devices take the boiler and turbine out of service in the event of failure of automatic desuperheaters, loss of circulating water, excessive turbine casing temperature, or loss of control air pressure during the starting process.

Generator and Transformer Differential. These electrical troubles are beyond the scope of this book.

Generator Grounds. A relay in the main circuit trips both turbine and boiler out of circuit when generator insulation fails.

Loss of Generator Fields. A relay in the field circuit trips the unit on any reduction of field current which would cause reactive flow into the generator and upset system stability. This relay functions also when out-of-step operation seems imminent.

Generator Motoring. A reverse-power relay in the high-pressure generator circuit trips out the unit when steam flow is not enough to supply no-load losses. Motoring of the unit is likely to damage the low-pressure blades by overheating from windage.

Auxiliary-circuit Faults. In case of faults in the individual auxiliary circuits relays disconnect them instantly. If faults should occur at the terminals of the auxiliary-supply transformers, suitable relays trip out the entire unit, generator and boiler.

Generator-feeder Faults. The unit system of this unit, from a protection viewpoint, extends from the fuel supply to and through the first bus section at a distant substation to which it connects. Hence the electrical protection covers the entire feeder, and, in case of a major fault in the feeder, the unit is tripped out.

Furnace-air-supply Failure. The furnaces of these boilers operate pressurized and have only f-d fans. In case all fans are lost by an interrupted power supply, relays trip the fuel supply as well as the turbine and generator. Complete closing of inlet and outlet dampers also trips the complete unit of service. The automatic combustion control reduces fuel input and generator load when the air supply cannot maintain a safe fuel-air ratio.

Excessive Furnace and Flue Pressures. If furnace pressure exceeds 30 in. H_2O, or the flue reaches 10 in. H_2O pressure, the entire unit trips out of service.

Complete Interruption of Fuel Supply. When fuel to the boiler is interrupted and the turbine not tripped, boiler pressure drops as steam continues flowing through turbine. A rapid drop in boiler pressure may cause a rise in drum level as water flashes to steam bubbles in the boiler-water circuits. This may cause carry-over into the turbine and damage the blading. Loss of fuel trips the entire unit.

CHAPTER 20

SUPERCRITICAL-PRESSURE STEAM STATIONS

20-1. Trend to High Pressures. Power-plant designers always strive to offset the effect of rising fuel prices on the cost of electrical energy. This can be done by improving the thermal efficiency, by raising the steam pressure and temperature, and by lowering the condensing pressure. Low condenser pressures have been realized for a long time, and we cannot expect much improvement beyond present practice.

High-steam-temperature application depends on the availability of metals that can withstand them. Alloys have been developed to hold 1050-F steam as normal commercial practice, but some developmental plants use temperatures up to 1150 F, and one plant will use 1200 F in the near future. The high cost of the alloys needed offsets somewhat the gain in efficiency realized.

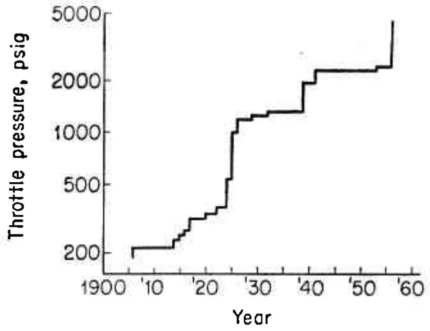


FIG. 20-1. Curve showing progress in increasing throttle steam pressure since 1910.

Higher pressures help to raise the average temperature at which heat is added to a steam power cycle, and this improves plant efficiency. Figure 20-1 shows the advance in steam pressures over the years.

The highest pressure that will be used is 5,000 psig in the most advanced *supercritical* or *superpressure* plants. Steam reaches its critical state at 3,206 psia and 705 F. To generate steam at supercritical conditions, the generator consists essentially of parallel tube circuits connected by an inlet and an outlet header. Pressurized feedwater enters the tube inlets to leave as superheated steam at the outlets. As the water absorbs heat energy, it gradually expands in volume with no internal bubbling or visible change in phase; it acts very much as an extremely dense gas.

20-2. Supercritical-pressure Cycle. Figure 20-2 shows a simple Rankine cycle with a supercritical pressure and temperature, and at ideal conditions. The feedwater pump pressurizes the water at *a* to a superpressure at *b* through an isentropic process. Heat addition from *b* to *c* produces a dense superheated steam. Isentropic expansion of steam in the turbine from *c* to *d* exhausts a wet steam to the condenser. From *d* to *a* the condenser removes the unavailable energy.

Figure 20-3 shows that superpressures raise thermal efficiency when the temperatures are above 900 F; this figure assumes an engine efficiency of 80 per cent.

Figure 20-4 shows the state relations for steam. There can be neither saturated water nor steam above the critical temperature of 705.4 F.

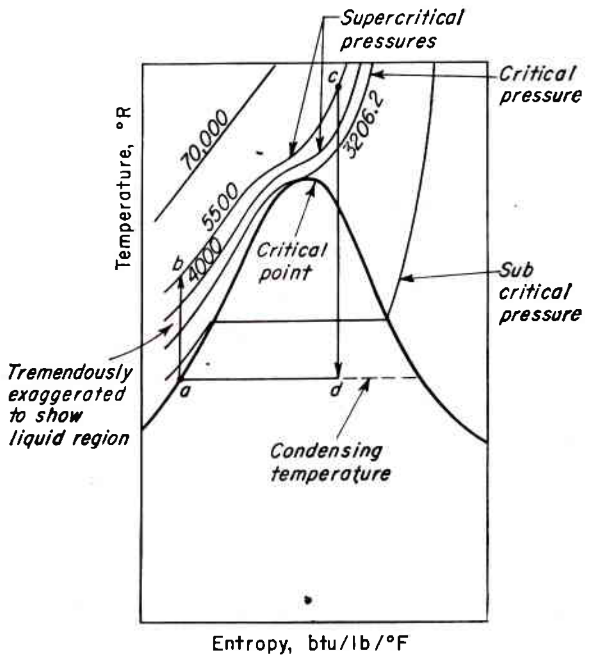


FIG. 20-2. Temperature-entropy analysis of simple supercritical cycle. Note that the diagram is not to scale and that the subcooled liquid region is tremendously magnified to represent supercritical-pressure lines.

Designers arbitrarily call superpressurized H_2O steam when its temperature is above 705.4 F and water when below this temperature. In the strict sense we cannot designate any part of the tube circuit in a superpressure steam generator as economizer, boiler, and superheater—designers, however, arbitrarily use these terms to indicate relative steam-water temperatures in the unit.

20-3. Steam-cycle Development. Early steam plants of the 1900s used simple cycles like cycle I (Fig. 20-5). Steam conditions were about 200 psia and 500 F. The *T-S* diagram shows the boiler input as the hatched area under line *CA*. Dashed line *AB* shows the steam expansion through the turbine at 80 per cent engine efficiency.

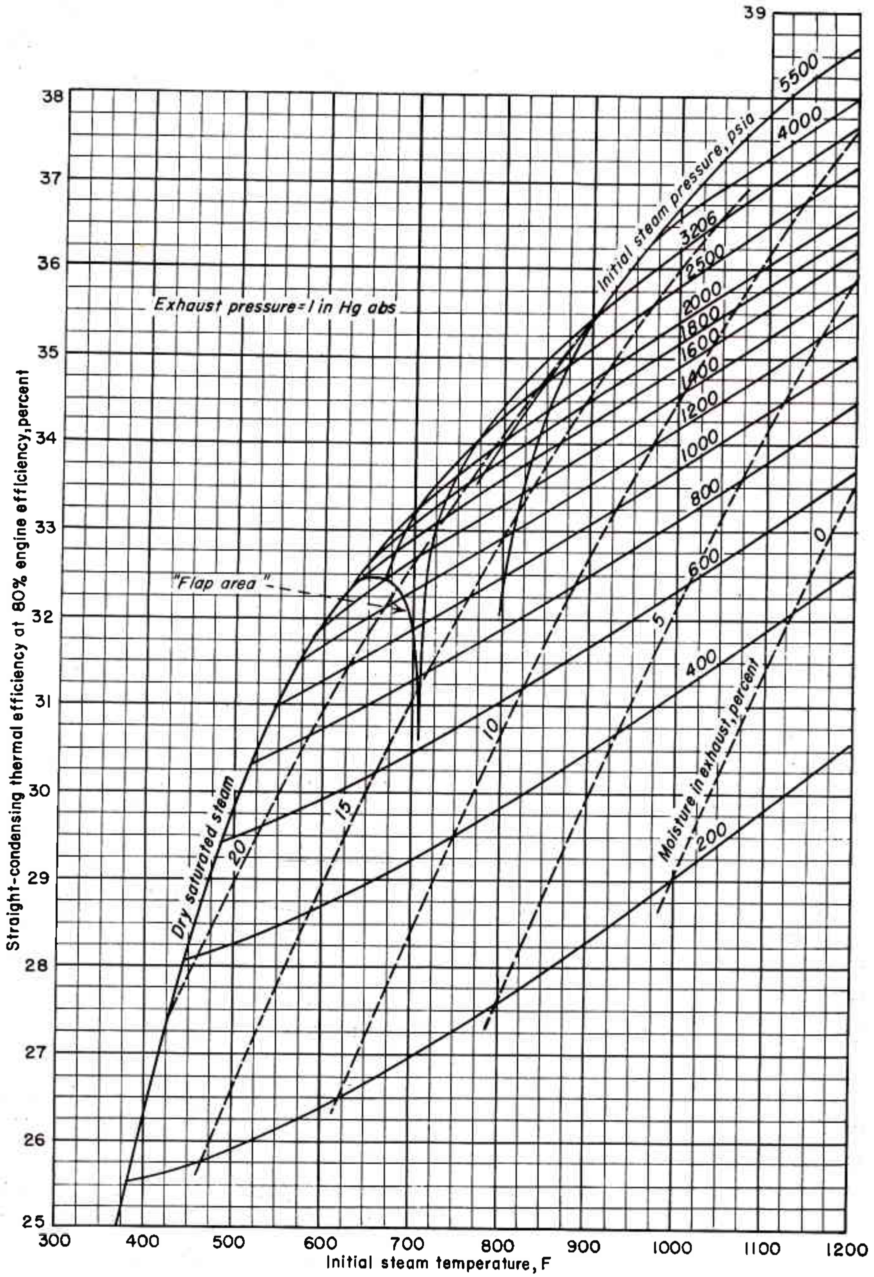


FIG. 20-3. Except for conditions near the critical (flap area) raising either steam temperature or pressure or both increases the thermal efficiency of a turbine. To obtain improved efficiency at superpressure, temperatures must be above 900 F.

The shaded area under *BC* measures the heat rejected in the condenser. The difference of the hatched and shaded area measures the work output of 1 lb of steam. The difference is not the remaining hatched area above *CB* but something less than this by the amount of the noncoincident area below and to the left of *B*. This develops with a turbine running at less than 100 per cent engine efficiency.

Table 20-1 summarizes important performance factors of this cycle and the other five to follow, assuming a boiler efficiency of 85 per cent and a turbine engine efficiency of 80 per cent.

Table 20-1. Comparison of Cycle Performance

	Cycle					
	I	II	III	IV	V	VI
Plant thermal efficiency, %	22.03	25.68	31.60	37.37	38.75	39.70
Plant heat rate, Btu per kwhr	15,500	13,300	10,800	9135	8810	8600
Plant throttle steam rate, lb per kwhr	10.78	7.25	6.35	8.43	6.79	5.95
Plant condenser steam rate, lb per kwhr	10.78	7.25	6.35	4.84	4.04	3.63
Throttle steam volume rate, cu ft per kwhr	29.38	34.53	1.156	1.534	1.236	1.084
First reheat steam volume rate, cu ft per kwhr					6.81	4.09
Second reheat steam volume rate, cu ft per kwhr						22.8
Condenser steam volume rate, cu ft per kwhr	6132	5490	3570	2720	2483	2450
Condenser heat rejection, Btu per kwhr	9740	6665	5710	4350	4020	3845
Exhaust steam moisture or superheat	13.7%	87 F	14.3%	14.3%	5.2%	22 F
Feed-pump work, % of plant net output	0.25	0.17	4.10	5.44	4.37	3.83

In cycle II (Fig. 20-6) steam temperature rises to 1150 F to improve performance in all respects. For constant steam velocity the front end of the turbine and the piping have to be larger because of the larger steam *volume* rate. The exhaust to the condenser is slightly superheated, but despite this the condenser can be made smaller because of the smaller volume rate of flow.

In cycle III the pressure rises to 4,500 psia and 1150 F to make a further gain in efficiency. Steam volume rates at throttle and condenser shrink markedly. Moisture in the exhaust rises to the danger level of 14.3 per cent. In addition to fuel-economy gains the higher steam conditions offer a marked reduction in size of the main plant components (Figs. 20-11 and 20-12). While more expensive metals will be needed in the high-temperature end of the cycle, such as superheater, piping, and first turbine stages, the size reduction in these parts helps offset the total cost.

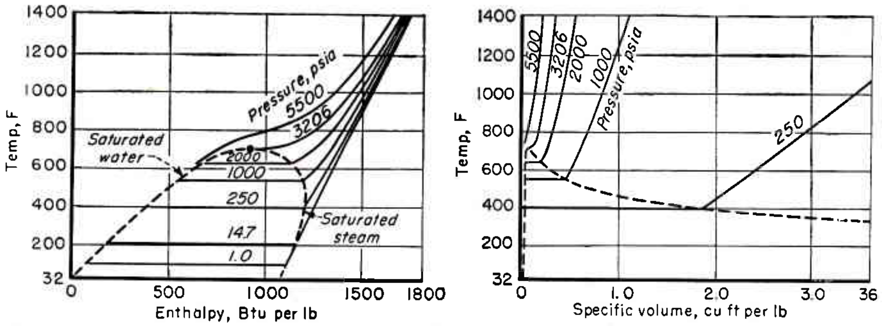


Fig. 20-4. As pressures increase, vaporization temperature increases but the enthalpy of vaporization decreases; also the density increases.

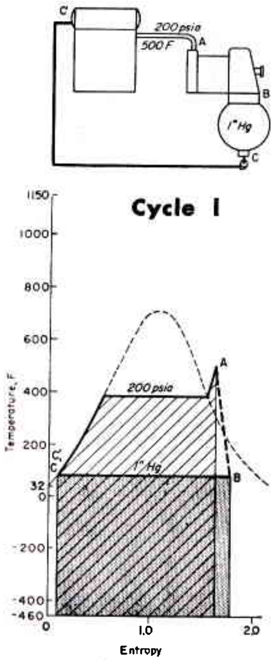


Fig. 20-5

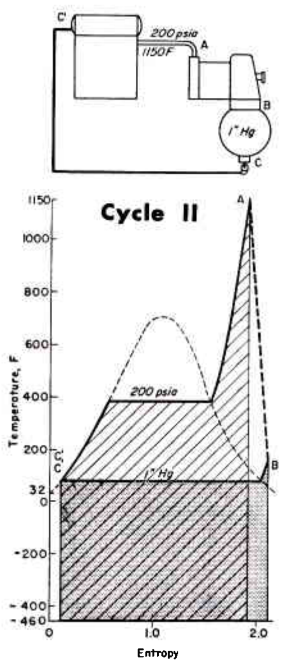


Fig. 20-6

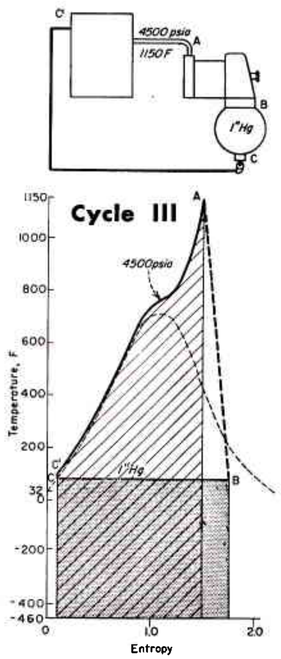


Fig. 20-7

Fig. 20-5. Cycle I, straight condensing cycle, operating at 200 psia and 500 F, expanding to 1 in. Hg abs back pressure at 80 per cent engine efficiency. Net work output is the difference between hatched and shaded areas.

Fig. 20-6. Cycle II, straight condensing cycle, operating at 200 psia and 1150 F, expanding to 1 in. Hg abs back pressure at 80 per cent engine efficiency.

Fig. 20-7. Cycle III, straight condensing cycle, operating at 4500 psia and 1150 F, expanding to 1 in. Hg abs back pressure at 80 per cent engine efficiency.

In cycle IV we add heat-recovery apparatus in the form of seven feedwater heaters. These reduce the heat input to each pound of steam to the hatched area under *AD* (Fig. 20-8). For each pound of steam entering the turbine throttle considerably less than 1 lb exhausts to the condenser because of the amount bled to the heaters. The heat rejected by 1 lb of steam in the exhaust is the area under *CB* (Fig. 20-8), but the shaded area is scaled down to represent the heat rejected in the condenser for each pound entering the throttle. Compared with cycle III, the heat rejected in cycle IV is proportionately more reduced than the heat input to the high-pressure steam; this markedly raises cycle efficiency. While the throttle steam rate rises, the condenser steam rate drops. The condenser volume-flow-rate decrease contributes to reducing the condenser's physical size.

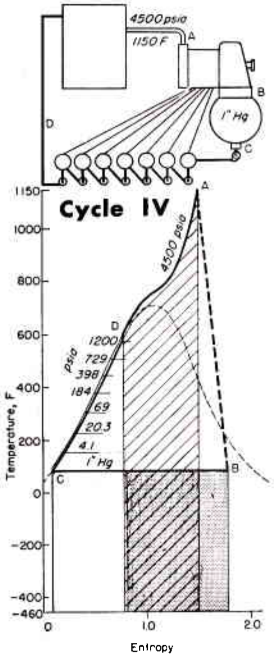


FIG. 20-8

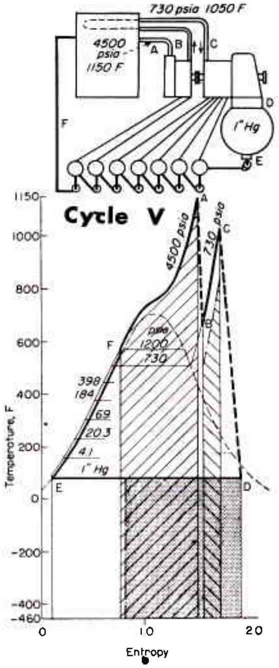


FIG. 20-9

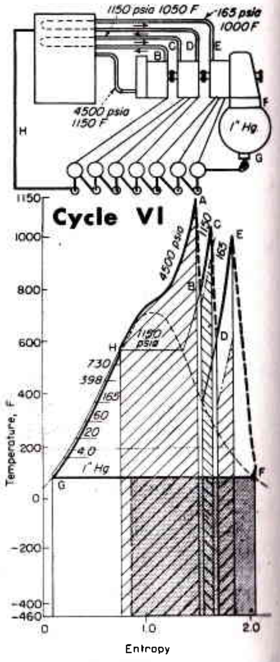


FIG. 20-10

FIG. 20-8. Cycle IV, regenerative feedwater heating cycle, operating at same pressure and temperature conditions as cycle III.

FIG. 20-9. Cycle V same as cycle IV, with one stage of reheating at 730 psia and 1050 F added.

FIG. 20-10. Cycle VI, regenerative feedwater heating cycle, with two stages of reheating and same number of heaters and throttle and exhaust conditions as cycle IV.

In cycle V (Fig. 20-9 and Table 20-1) we add one stage of reheating at 730 psia back to 1050 F, the commercial level. The area under *BC* measures the reheater input to 1 lb of steam, but since some of the steam is bled for feedwater heating, less than 1 lb goes to the reheater for each pound of throttle steam. The reduced hatched area shows the amount of reheat for each pound of steam passing through the throttle. The sum of the hatched area, less the gray area, measures the net heat output of the cycle per pound of steam entering the throttle. The reheating makes gains in all performance factors, justifying the expense of the additional piping and reheater in the steam generator.

Cycle VI (Fig. 20-10, Table 20-1) uses two stages of reheating, the first at 1,150 psia to 1050 F and the second at 165 psia to 1000 F. This gives another performance improvement all around. The exhaust steam to the condenser is slightly superheated. The sum of the hatched areas minus the gray area measures the output of the cycle.

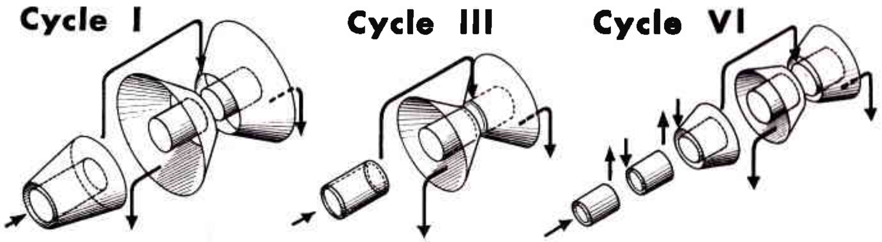


FIG. 20-11. As pressures and temperatures increase, the turbine steam-path cross-sectional area decreases. This decreases the physical size of the over-all turbine for the same plant capacity.

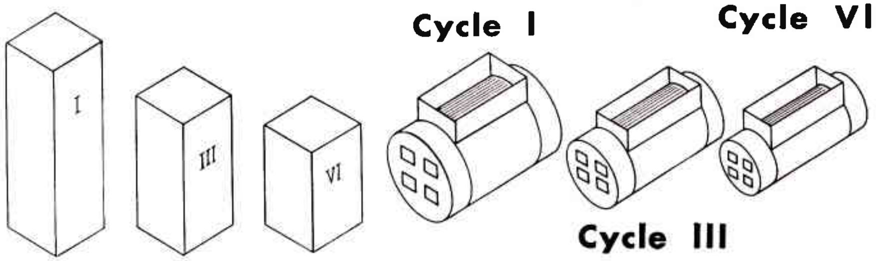


FIG. 20-12. Higher pressures and temperatures and improved heat utilization reduce furnace volume (left) and condenser size (right) for the same capacity of steam plant.

Figure 20-11 shows the comparative sizes of the turbine steam paths needed in cycles I, III, and VI. Single-double-flow tandem-compound was the arrangement used on the first unit at Philo Station, with 125 mw capacity. Since the steam path primarily dictates the turbine over-all size, cycle VI would use relatively small units. The high-pressure and intermediate-pressure turbines would be relatively small. The first-stage turbine inlet as drawn is much too large. Blades for a 120-mw turbine would only be a fraction of an inch high to accommodate the high-density steam.

Figure 20-12 shows the relative sizes of furnace volume and of main

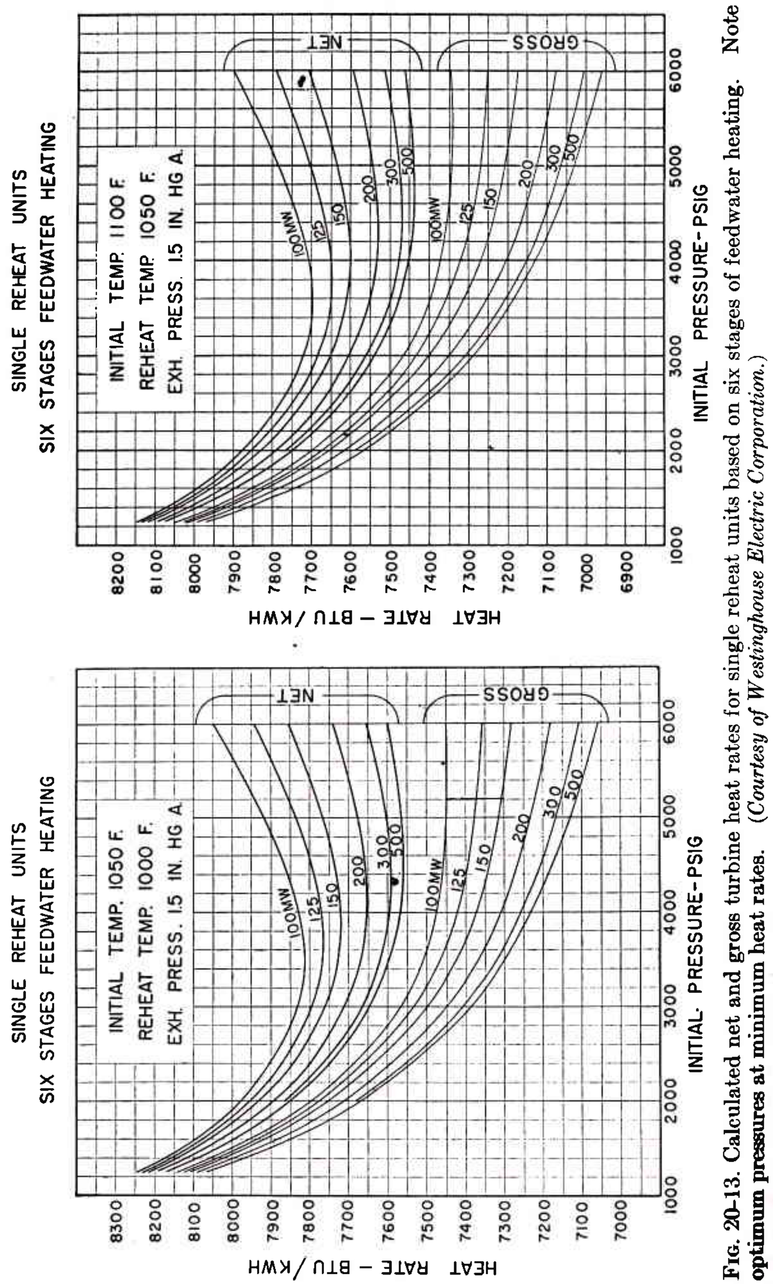


Fig. 20-13. Calculated net and gross turbine heat rates for single reheat units based on six stages of feedwater heating. Note optimum pressures at minimum heat rates. (*Courtesy of Westinghouse Electric Corporation.*)

condensers needed for cycles I, III, and VI. Here also we find that high steam conditions and heat-recovery cycles contribute toward reducing component physical size.

20-4. Performance Trends. Performance expressed as a heat rate in Btu per kilowatthour can be determined for: (1) turbine alone—gross, (2) turbine output less feed-pump input—net, or (3) over-all performance of fuel input and net station output—station or plant.

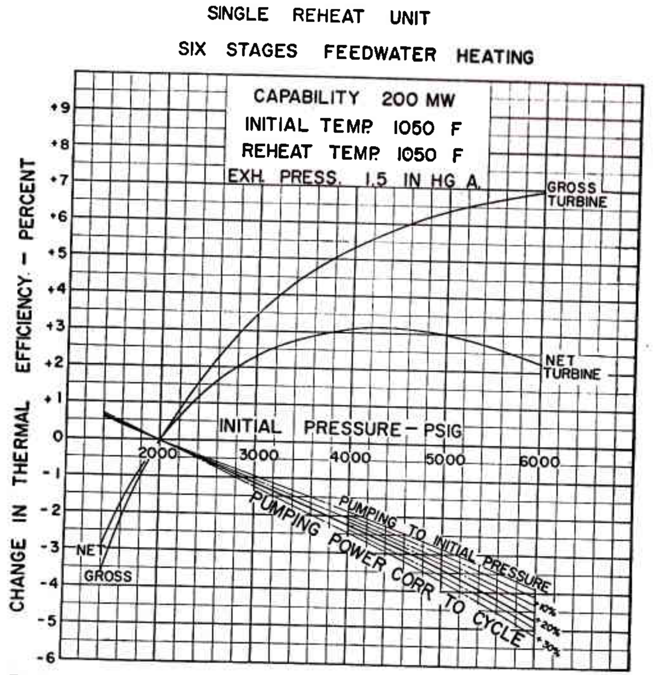


Fig. 20-14. Thermal-efficiency change due to pumping power. Note that the values represent changes from a base of 2,000 psig, 1050 F, reheat to 1050 F.

Figure 20-13 shows expected performance for a variety of turbine ratings at different pressures and temperatures. The turbine performance is given as *gross* or *net,* the difference showing the effect of pumping power needed.

The condensate pump discharges feedwater to three closed heaters followed by an open heater. The boiler feed pump takes suction from the open heater and discharges through two high-pressure closed heaters to the steam generator. Note the effect of pumping power in producing an optimum throttle pressure for best performance (lowest heat rate).

Figure 20-14 also shows the effect of pumping-power needs. Feedwater pumps discharge at pressures greater than throttle pressure to overcome line, superheater, and control-valve resistances; the curves

TABLE 20-2. SUPERCRITICAL-PRESSURE PLANT EQUIPMENT AND DATA

	Utility company					
	Ohio Power Company	Philadelphia Electric Company		Cleveland Electric Illuminating Company	Indiana & Michigan Electric Company	Ohio Power and Appalachian Electric Power Companies
Plant—Unit No.	Philo—6	Eddystone—1	Eddystone—2	Avon—8	Breed—1	Philip Sporn—5
Plant location	Philo, Ohio	Eddystone, Pa.	Eddystone, Pa.	Avon Lake, Ohio	Fairbanks, Ind.	New Haven, W.Va.
First operation	March, 1957	December, 1958	January, 1960	Early 1959	July, 1959	December, 1959
Capability, kw	125,000	325,000	325,000	250,000	450,000	450,000
Pressure, psig:						
Throttle	4,500	5,000	3,500	3,500	3,500	3,500
1st reheat	1,143	1,028	936	645	1,000	1,000
2d reheat	192	236	314	None	300	300
Temperature, °F:						
Throttle	1150	1200	1050	1100	1050	1050
1st reheat	1050	1050	1050	1050	1050	1050
2d reheat	1000	1050	1050	None	1050	1050
Boiler data:						
Manufacturer	Babcock & Wilcox	Combustion Engrg.	Combustion Engrg.	Combustion Engrg.	Babcock & Wilcox	Babcock & Wilcox
Capacity, lb per hr	675,000	2,000,000	2,178,000	1,715,000	2,923,000	2,923,000
Furnace heat release, Btu per sq ft per hr	....	87,000	89,000	85,700		
Firing method	Cyclone furnace	Pulverized coal	Pulverized coal	Pulverized coal	Cyclone furnace	Pulverized coal
Furnace layout	Single	Twin	Twin	Twin	Single	Single
Furnace bottom	Slag tap	Dry bottom	Dry bottom	Dry bottom	Slag tap	Dry bottom
Air-heater type	Tubular	Regenerative	Regenerative	Regenerative	Regenerative	Regenerative
F-d fans, drive	2 motor-driven	4 motor-driven	4 motor-driven	2 motor-driven	3 motor-driven	2 motor-driven
I-d fans, drive	2 motor-driven	4 motor-driven	4 motor-driven	2 motor-driven	None	None
Boiler feed pumps	2 motor-driven	4 motor-driven 2 turbine-driven	4 motor-driven 2 turbine-driven	1 motor-driven 2 turbine-driven	2 motor-driven 1 turbine-driven	2 motor-driven 1 turbine-driven
Pump manufacturer	DeLaval Stm. Turb.	Ingersoll-Rand	Ingersoll-Rand	Worthington	Ingersoll-Rand	Ingersoll-Rand
Pump-heater system[a]	5H-P-2H-P	4H-P-5H-P-P[b]	4H-P-5H-P-P[b]	4H-P[b]-P[b]-4H[c]	5H-P-2H-P[b]-2H	5H-P-2H-P[b]-2H
Turbine builder	General Electric	Westinghouse	General Electric	Westinghouse	General Electric	General Electric
Turbine type	TC-2E	CC-2E	CC-6E	TC-3E	CC-6E	CC-6E
Turbine speed, rpm	3,600	3,600/1,800	3,600/3,600	3,600	3,600/3,600	3,600/3,600
No. bleed points	5	8	7	5	8	8
No. heaters	7	2 lines, 9 each	2 lines, 8 each	8	9	9
Deaerating pressure, psia	210	54	76	118	48	48
Condenser, sq ft	55,000	150,000	150,000	113,000	198,000	198,000
Condenser builder	Allis-Chalmers	Westinghouse	Westinghouse	Allis-Chalmers	Allis-Chalmers	Allis-Chalmers
High-temp piping	Type 347	Type 316	....	Type 316	Type P22	Type P22
Plant net heat rate, Btu per kwhr	8954	8016	8250	8603	8450	8450

[a] Pump-heater system listed from condensate pump to boiler inlet.

[b] Turbine-driven pump.

[c] Two strings of 4 high-pressure heaters.

CC, cross compound; E, exhausts; H, heater; P, pump; TC, tandem compound.

give corrections for these excess pressures needed in an actual cycle. These are based on 75 per cent pump efficiency and 89 per cent combined efficiency for motor, transformer, gear, and hydraulic coupling.

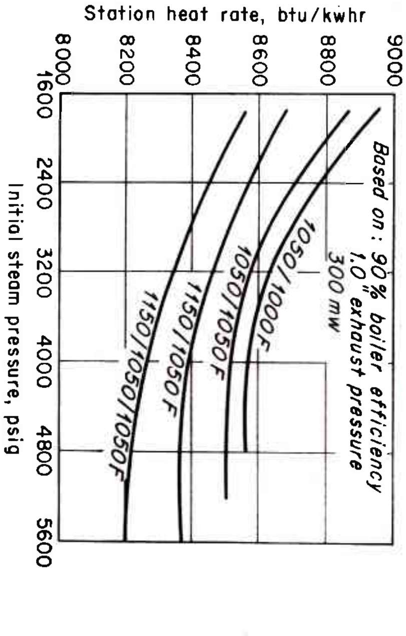


FIG. 20-15. Typical anticipated station heat rates for several steam conditions. (*Courtesy of Allis-Chalmers Manufacturing Company.*)

Figure 20-15 summarizes possible station heat rates for a range of throttle steam pressures and temperatures with modern feedwater heating arrangements. Optimum heat rate alone does not determine an economic station choice; the costs of station equipment must be considered with fuel costs to arrive at the best plant.

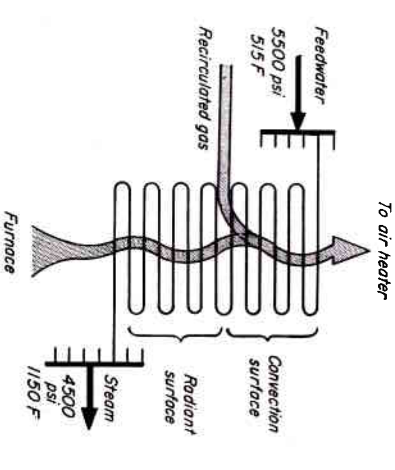


FIG. 20-16. Schematic circuit of steam generator shows simplicity of design and once-through flow of water to emerge as steam. No recirculation of water within unit as in conventional steam boiler. Recirculating part of flue gas helps to prevent ash slagging of convection surface.

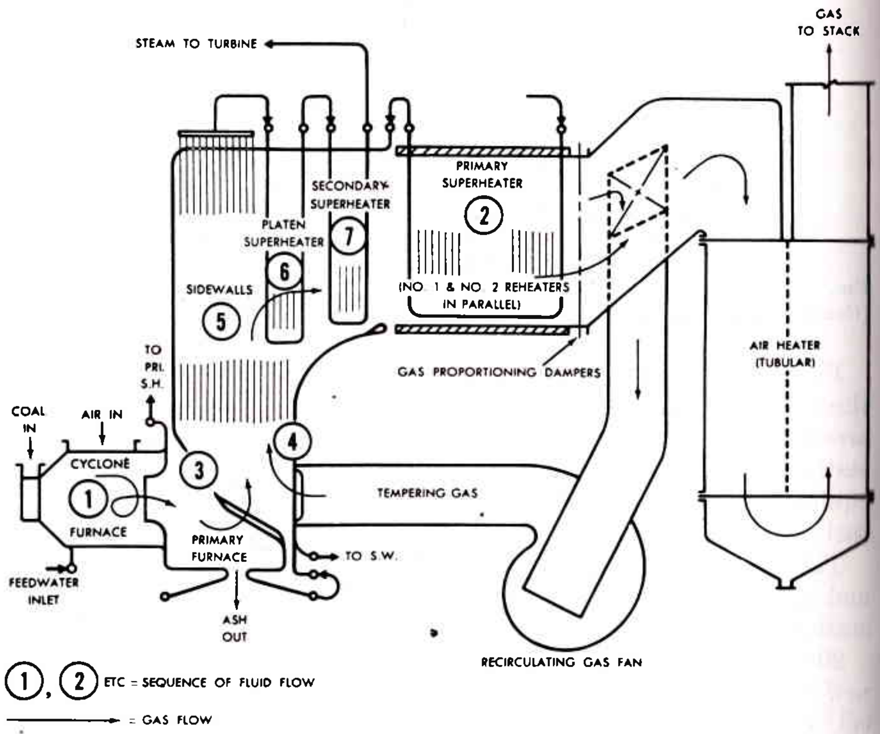
Table 20-2 lists data for existing and proposed supercritical-pressure stations.

20-5. Plant Components. The new conditions met in a supercritical plant raise problems different from those in a conventional subcritical station. Let us consider what this does to the components.

Steam Generator. Since there is no bubbling of steam in the water absorbing heat and the density of the "steam" and "water" are nearly equal, we cannot use natural circulation as in conventional boilers. We use *once-through* steam generators as described previously. These have been used for subcritical steam for many years in Europe in the Benson, Sulzer, and

Ramsin boilers. This type of boiler needs no large drum but uses small-diameter thin-walled tubes, a means of saving steel.

Figure 20-16 shows a once-through steam generator in schematic form. Figure 20-17*a* shows the outline of the first superpressure unit in the United States, Philo Unit 6. This steam generator produces 675,000 lb per hr steam at 4,500 psia and 1150 F. Steam is reheated in two stages, the first to 1050 F at 1,150 psi and the second to 1000 F at 180 psia. Figure 20-17*b* shows a section through a model of the unit.

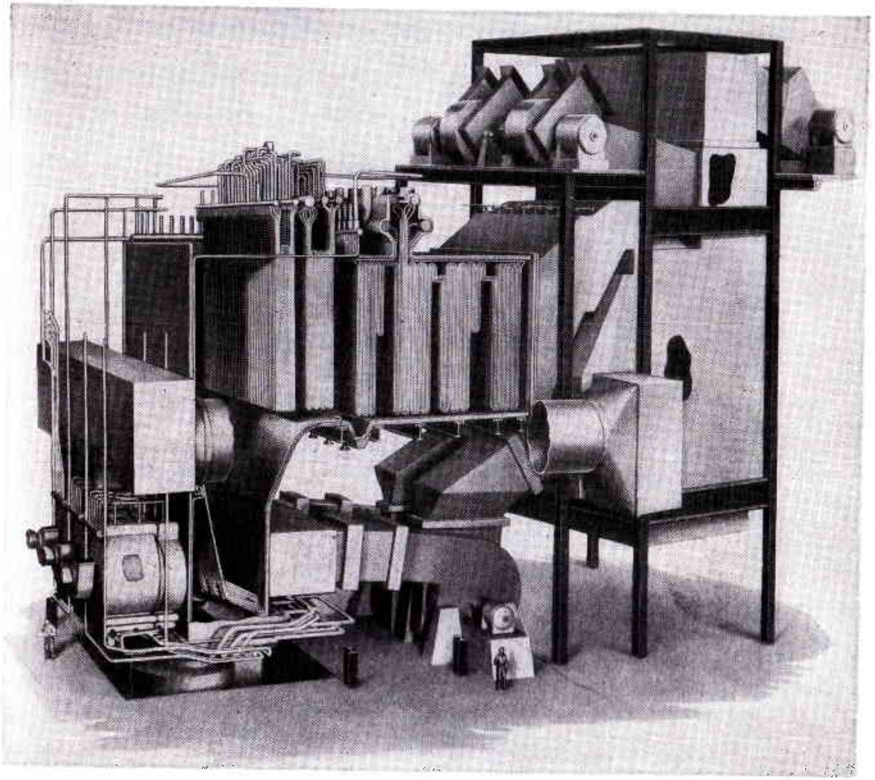


(*a*)

Three cyclone furnaces fire the unit. The recirculating fan draws off some of the combustion gas before the tubular air heater and reinjects it into the secondary furnace to temper the temperature of the combustion gas before it enters the platen superheater. This cools the molten ash so that it will not build up as slag on the tubes.

On the fluid side feedwater first flows through the wall tubes of the cyclone furnaces in series. It then passes through the primary superheater tubes and then to the primary furnace wall tubes; remember that there are a number of tube circuits in parallel. From the wall tubes the steam passes through the primary and secondary superheaters before leaving for the turbine.

The fluid-flow path consists of 19 circuits in parallel. The outlet manifold distributes the completely mixed steam to four turbine leads. A turbine bypass system diverts steam flow around the turbine to the condenser until the steam is warm enough for rolling the turbine during starting. These boilers may also be used for generating subcritical steam at varying pressures.



(*b*)

Fig. 20-17. (*a*) Simplified side elevation for fluid- and gas-flow circuits of Philo steam generator No. 6; (*b*) sectional model. (*Courtesy of The Babcock & Wilcox Company.*)

Figure 20-18 shows a later design of a once-through steam generator to supply 2,900,000 lb per hr steam to a 450-mw turbine.

Steam Turbine. Superpressures prove practical for large-capacity units since the steam is very dense. This requires relatively small first-stage flow areas. Unless the total flow is large enough, the buckets and nozzles may be so small that serious frictional losses may make the turbine impractical by offsetting the gains offered by supercritical conditions.

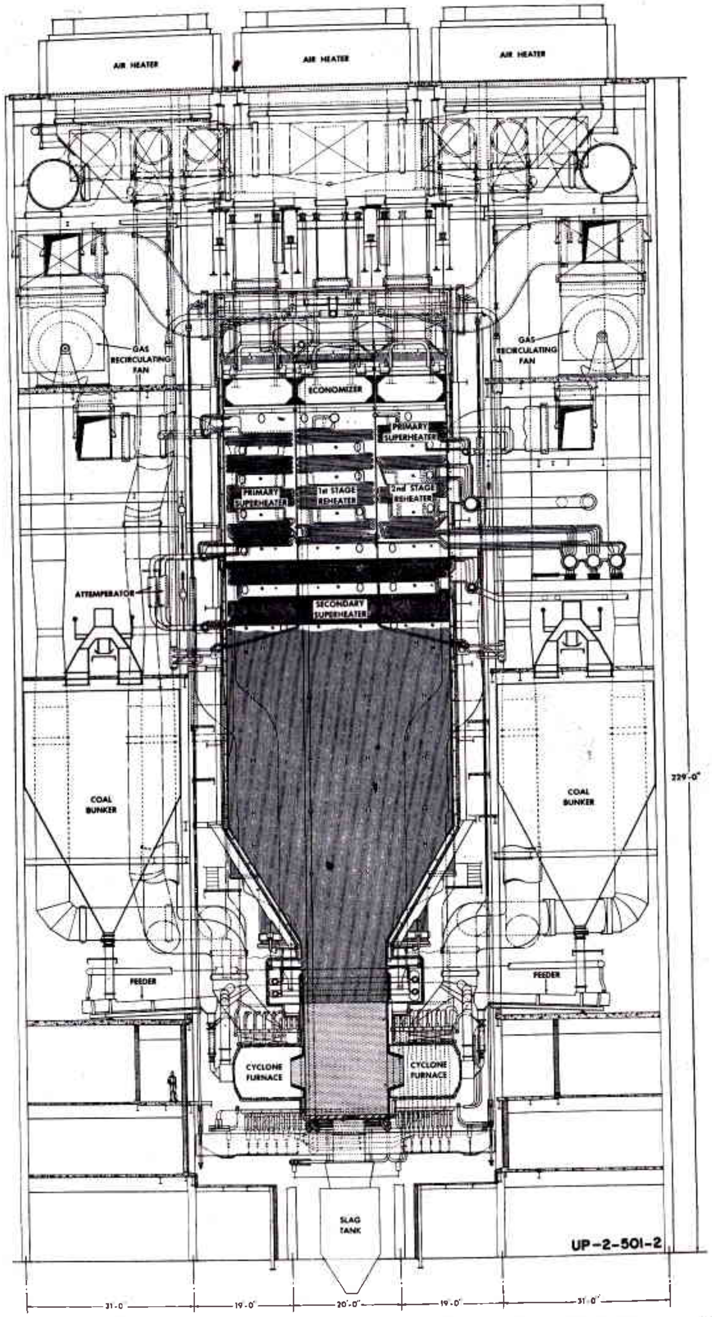


Fig. 20-18. Universal pressure steam generator for Breed Plant Unit No. 1. (*Courtesy of The Babcock & Wilcox Company.*)

Designers now consider units up to 750 mw as practical, but they will depend on improving rotor forgings. With these large sizes and advanced steam conditions heat rates of 7500 Btu per kwhr are forecast.

Temperatures above 1050 F need austenitic materials for safe operation, but they are expensive. Designers strive to minimize the amount needed in superpressure equipment. Single-shaft turbines reduce the building volume needed, but as capacities grow, the exhaust annulus needed to handle the high volume flow of the turbine exhaust makes it necessary to adopt multiple arrangements using more than one shaft. Figure 20-19 shows the turbine arrangement used in the Breed Plant of the American Electric Power System, which will produce 450 mw.

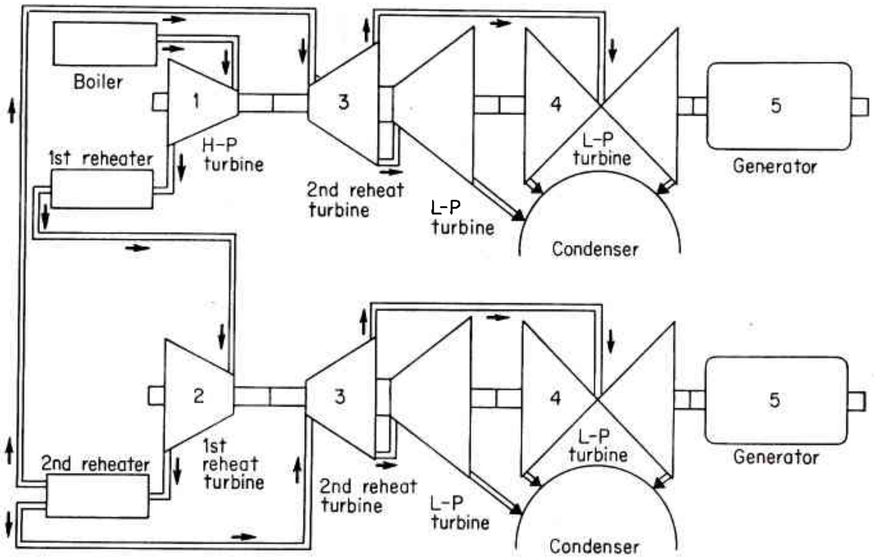


Fig. 20-19. Schematic diagram of 450-mw supercritical-pressure turbine generator for Breed Plant. Generators are rated at 265,000 kva each at 16 kv. Speeds of both units are 3,600 rpm. Note sextuple exhaust. See Table 20-2 for other data. (*Courtesy of American Electric Power Service Corporation.*)

20-6. Water Problems. With no separation of phases of steam and water as in a conventional boiler, the once-through boiler presents problems of leaving solids in the tubes as the water changes to steam. Water contaminants such as silica, sodium chloride, sodium sulfate, and calcium sulfate have greater solubility in supercritical steam compared with subcritical steam. This tends to keep solids in solution in supercritical steam generators and to reduce the solids deposit on the tube interiors. This, however, makes silica deposits in the turbine more likely as the steam expands.

Experience as of this writing shows that supercritical water tends to

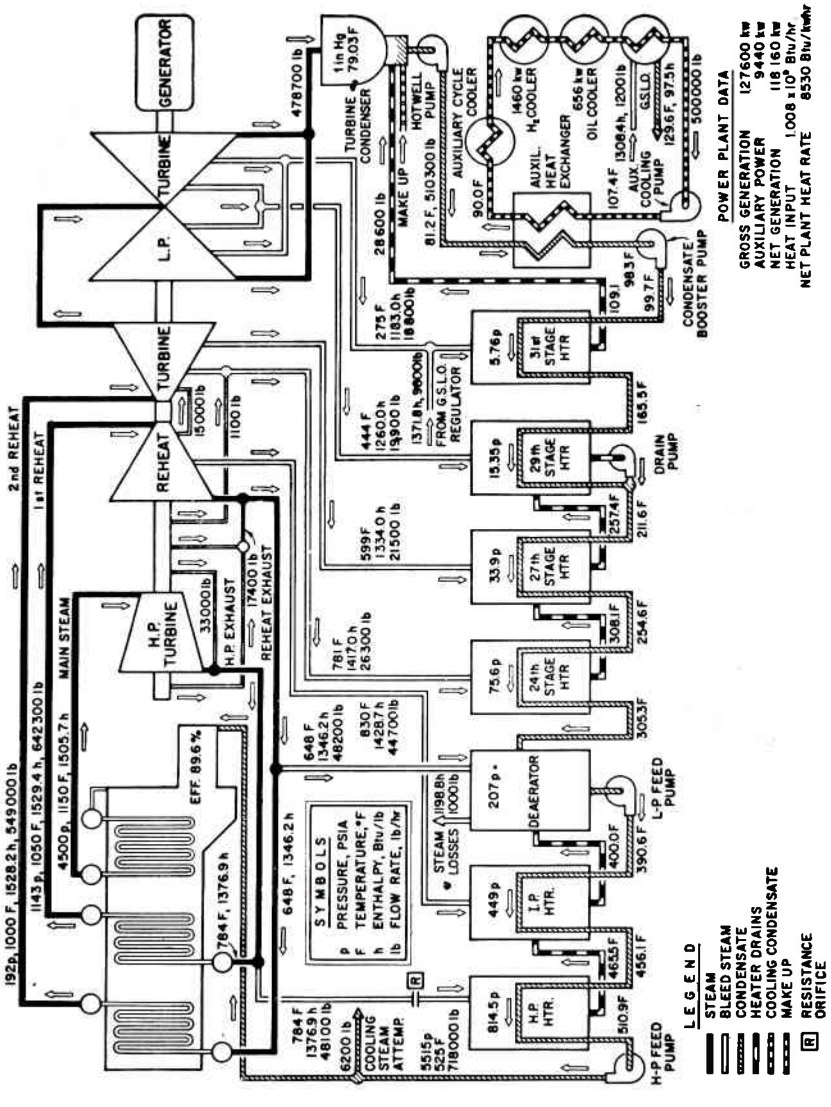


Fig. 20-20. Full-load heat cycle of experimental superpressure Philo Unit No. 6. (*Courtesy of American Electric Power Service Corporation.*)

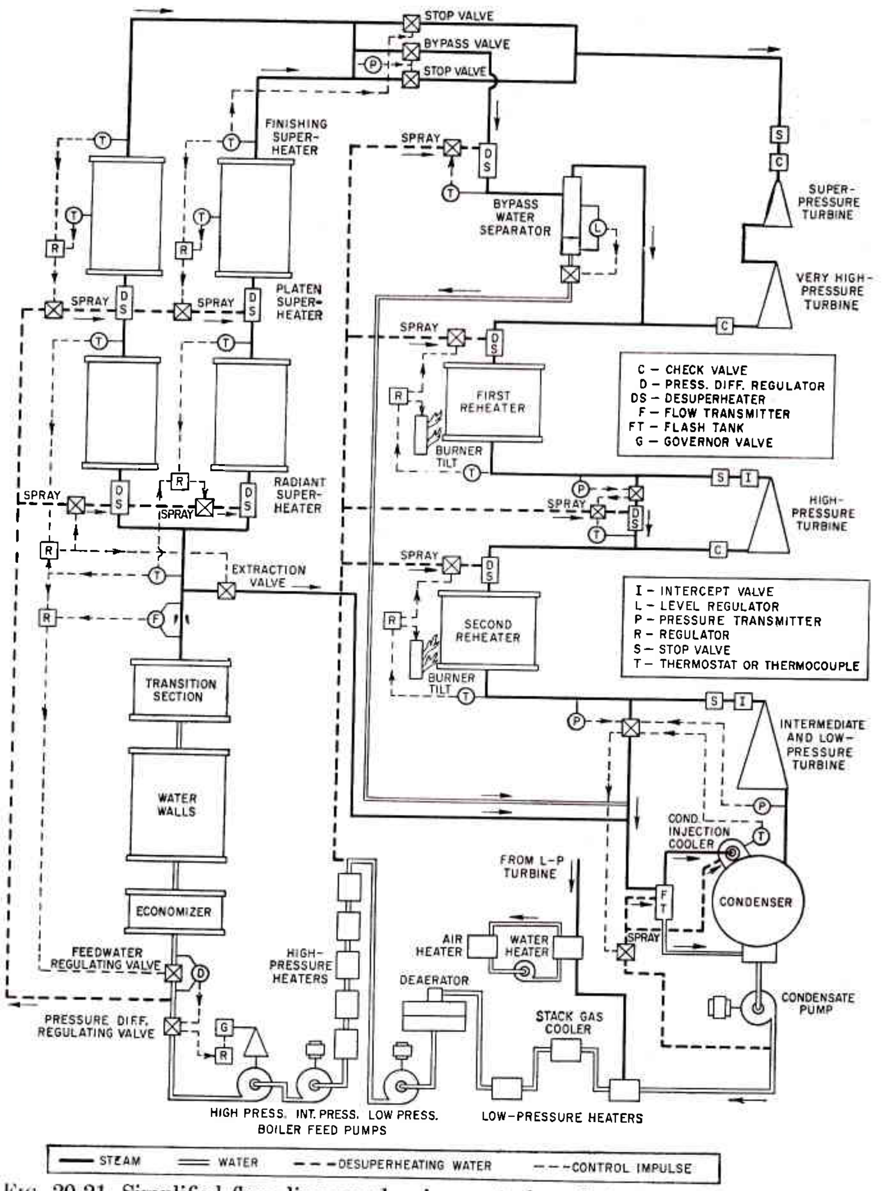


Fig. 20-21. Simplified flow diagram showing controls and start-up system for Eddystone Unit No. 1. Components of boiler are indicated by rectangular blocks at left, and turbine components are shown by triangular blocks at right. (*Courtesy of Philadelphia Electric Company.*)

dissolve oxides of iron and deposit them on the generator heating surfaces. Superpressure water tends to hold solutions of oxides of iron, chromium, nickel, and metal alloys. These may deposit in the turbine as the fluid becomes subcritical in the turbine or may be carried into the condenser condensate. It appears that, if metal oxides are put into the steam generator in the feedwater, they may go back into solution. A steady concentration may make the oxides dissolve under steam-generator conditions and precipitate in the turbine flow path.

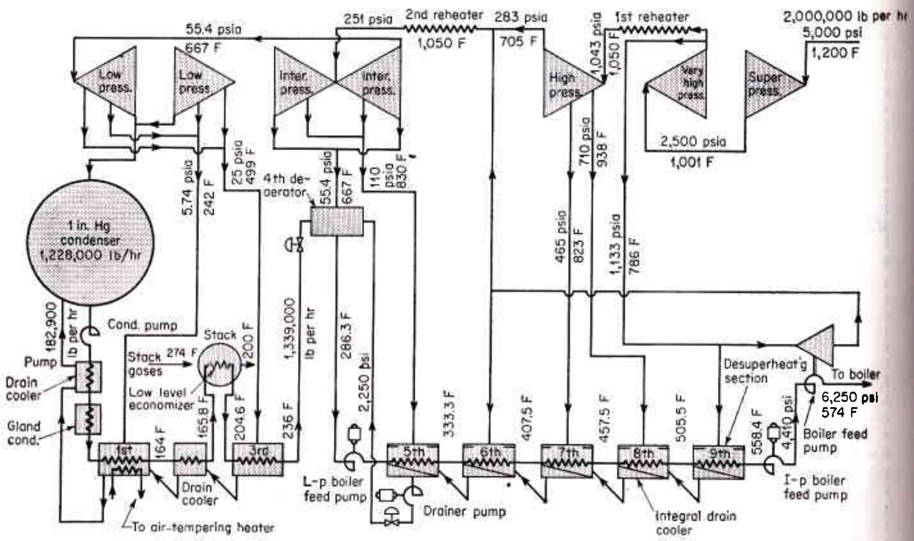


FIG. 20-22. Schematic arrangement of feedwater heater cycle for Eddystone Unit No. 1. Feedwater at 574 F is provided by nine feedwater heating stages. (*Courtesy of Philadelphia Electric Company.*)

It seems that, since there is no vaporization in supercritical units, no steam blanketing can form and burn out tubes. Dissolved solids tend to concentrate on the inner tube surface, where the fluid turns to steam. These deposits may attack the steel. But since no clear-cut vaporization takes place in supercritical generators, it seems unlikely that such troubles will develop.

20-7. Station Layouts. Figure 20-20 shows the heat balance for Philo Unit 6, the first superpressure plant. Its steam conditions were chosen because theoretically they give the optimum value of efficiency. In addition the operating pressures of the feedwater lines, boiler, and steam lines lie within the present codes. Seven feedwater heating stages and two reheats are used in this cycle.

Figure 20-21 shows the cycle for the Eddystone Unit 1 with a throttle pressure of 5,000 psia and 1200 F temperature, ultimately. The designers chose the most advanced conditions that might be economically feasible. The station uses nine feedwater heaters and two reheats. The cycle uses the lowest-pressure bleed steam to temper combustion air and a low-level economizer to heat condensate by lowering stack temperature to about 200 F (see Fig. 20-22).

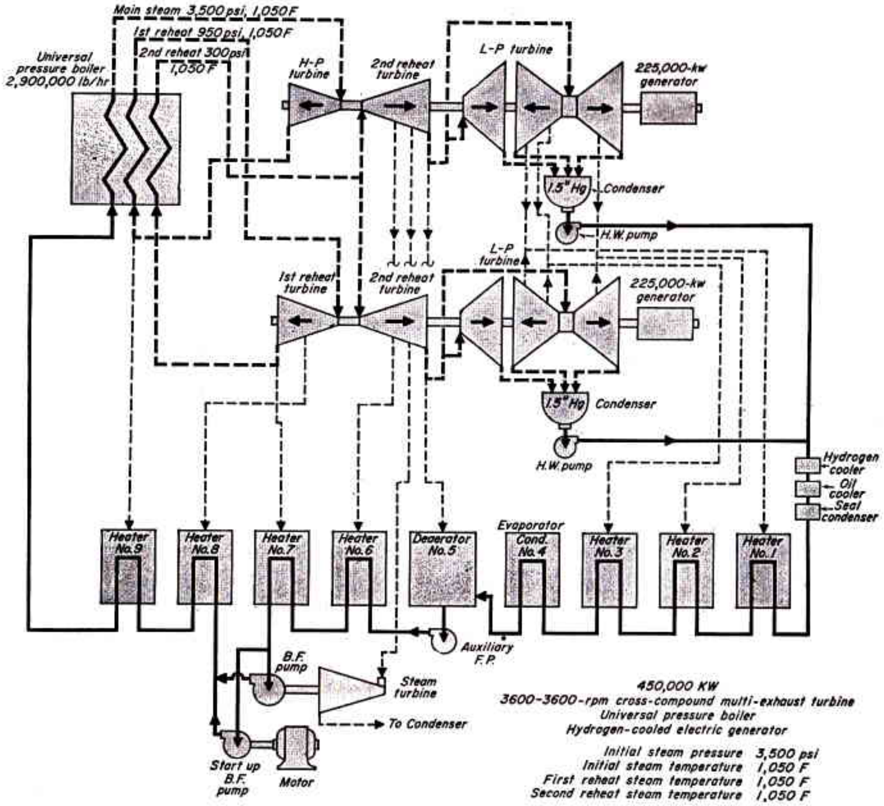


FIG. 20-23. Preliminary steam cycle of commercial supercritical unit for Breed Plant. (*Courtesy of American Electric Power Service Corporation.*)

Figure 20-23 shows the preliminary cycle for the Breed Plant with steam conditions of 3,500 psig, 1050 F, and double reheat to 1050 F as the economic conditions. The design of the 450-mw unit was undertaken by the American Electric Power Corporation after completing the Philo Unit 6.

PROBLEMS

20-1. Check the thermal efficiency shown in Fig. 20-3 under the following conditions: (*a*) 5,500 psia, 1150 F; (*b*) 4,000 psia, 1050 F.

20-2. Check the data for cycles I to VI in Table 20-1.

20-3. Check the consistency of the data in Fig. 20-14 at 5,000 psia.

20-4. Refer to Fig. 20-20 and (*a*) check the heat input; (*b*) calculate the coal used in tons per hour; (*c*) check the heat balance for each heater.

20-5. Refer to Fig. 20-22, and (*a*) calculate stack gas flow through the low-level economizer in pounds per hour; (*b*) calculate steam bled to deaerator from the intermediate-pressure turbine, assuming 3 per cent boiler blowdown and 5 F subcooling in heater No. 5; (*c*) calculate the coupling work of the superpressure turbine based on 95 per cent mechanical efficiency.

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CHAPTER 21

NUCLEAR POWER STATIONS

21-1. Nuclear Fuels. Scientists have long theorized that considerable energy could be released by fissioning (breaking up or splitting) heavy atoms. In the late 1930s this had been done on a minute laboratory scale, and the impetus of World War II brought practical realization in the form of the atom bomb. The enormous release of energy from a relatively small mass of material makes this method of particular interest to central-station energy generation.

Certain heavy atoms are in a precarious state of internal balance, to the extent that by inducing a neutron to enter their nuclei they will fission into two smaller nuclei, eject two or three neutrons, and the fragments emit a gamut of rays: alpha, beta, and gamma. The energy comes from a minute fraction of the original mass converting according to Einstein's famous law: $E = mc^2$, where E represents energy, m mass, and c the speed of light.

This equation tells us that 1 pound of mass equals 11.3 billion kilowatthours of energy, or 38,660 billion Btu. Most of this energy in fission appears as internal energy of the fission fragments, while the kinetic energy of the neutrons and the radiations quickly converts to internal energy in the materials that absorb them. (Remember that internal energy is mainly the kinetic motion of atoms and molecules.)

Einstein's law also governs the release of energy by combustion. But this chemical process involves only the disappearance of mass of the electrons associated with the atoms. In nuclear fuels the energy comes from disappearance of mass from the nuclei of the atoms.

Any atom can be fissioned in an accelerator or similar device for energy release. But in a practical fuel we can do this only when it can sustain a *chain reaction.* This means that when an atom fissions, the neutrons ejected will be absorbed by other heavy atoms to fission in turn. When this condition prevails, a mass of nuclear fuel can be used as a source of heat energy.

The only natural substance that will function as a nuclear fuel is uranium. In pure form it is a heavy, hard nickel-white metal—it oxidizes in air and water. Uranium is widely distributed as an oxide ore in concentrations of less than 0.3 per cent throughout the world.

The richest deposits lie in the Belgian Congo, northern Canada, and the Colorado Plateau in the United States. It can be recovered as a by-product from some gold-ore and phosphate processes.

Natural uranium wherever found consists of:

Uranium 238	99.3%
Uranium 235	0.7%
Uranium 234	Trace

Of these three natural isotopes only uranium 235 (U-235) will fission in a chain reaction. A small portion of the U-238 will also fission when absorbing a neutron, but it cannot sustain a chain reaction.

21-2. Man-made Fuels. Uranium 235 in natural uranium fissions more readily when the neutron it absorbs moves relatively slowly, at a speed comparable with that of a "hot," or thermal atom. A neutron at 60 F travels at about 4,830 mph, and the speed varies with the square root of the absolute temperature. Neutrons emitted by fissioning U-235 fragments are *fast neutrons* and have about 1,000 times the energy of a *thermal neutron* most suited for absorption by U-235.

To sustain a chain reaction in natural uranium, the fast neutrons must be moderated (slowed down) to thermal speeds. This is done by letting neutrons collide several hundred times with nuclei of other light atoms, such as hydrogen, helium, beryllium, or carbon. When a thermalized neutron passes close to a U-235 nucleus, the short-range attractive forces have time to pull the neutron into the nucleus and initiate fission.

To avoid using a moderator, uranium fuels may be "enriched" by increasing the concentration of U-235 from as little as 1 per cent to as much as 95 per cent and more. Highly enriched uranium will sustain a chain reaction without a moderator, i.e., with fast neutrons. Enriched fuels are more costly because of the processing needed.

While natural uranium contains only 0.7 per cent fissionable material in the form of U-235, a fortuitous process takes place with U-238 that converts it to a fissionable material also. Some of the excess of neutrons released by a chain reaction in the U-235 isotopes are absorbed by the U-238 atoms. This initiates a series of processes with a half-life of less than 2½ days. During these processes the U-238 atom (which becomes a U-239 atom with the absorbed neutron) emits a gamma ray and two beta particles, becomes neptunium 239 (Np-239) for a while, and ultimately transforms into plutonium 239 (Pu-239), a toxic, but highly fissionable material. Plutonium 239 exists in nature in only trace amounts.

If the chain reaction and released neutrons can be controlled, this means that all of natural uranium can potentially become fissionable material, i.e., a nuclear fuel. This is called breeding and ideally could be carried on while natural uranium reacts to produce heat. In the reactors running today part of the U-238 transforms to Pu-239, which then fissions and contributes to heat production.

Thorium 232 (Th-232), a natural material, can also be converted to a fissionable material by placing it close to U-235, sustaining a chain reaction. By absorbing an excess neutron Th-232 changes to protactinium 233 (Pa-233) and ultimately to U-233 during a process with a half-life of about 27½ days and during which it emits a gamma ray and two beta particles. Uranium 233 can be fissioned by either fast or thermal neutrons and can sustain a chain reaction like U-235 and Pu-239. Uranium 238 and thorium 232 are called *fertile materials.*

21-3. Nuclear Reactor. To control the "burning" of fissionable fuels, they must be placed in a reactor which largely functions to control the release and absorption of neutrons during a chain reaction. Neutrons released in a chain reaction are disposed of in four different ways: (1) escape from the fissionable material; (2) nonfission capture by U-238 to form Pu-239; (3) nonfission capture by control-rod material, moderator nuclei, fission fragments, and impurities; and (4) fission capture by U-235, Pu-239, and U-233.

A chain reaction producing a constant rate of heat energy can continue only if the neutrons released by fission just balance the four types of disposal listed above. If neutron production rate drops below disposal rate, the chain reaction stops; if it exceeds disposal rate, fissioning rate increases and may get out of control. A reactor is of *critical size* when the fission-produced neutrons just balance those disposed of in the four categories. For a practical reactor the fissionable fuel in it must be larger than the critical size.

To regulate fissioning (heat production) in a reactor, control rods of boron or hafnium, neutron-absorbing materials, are inserted into the fuel. To increase the fissioning rate of a chain reaction, the control-rod penetration is reduced a small fraction of an inch so that the control rods absorb fewer neutrons, which are then available to fission more atoms in a given time. This allows the fissioning rate to increase. When the new fissioning rate has been established, the control rods are again reinserted into the fuel to absorb the excess neutrons and keep the chain reaction from growing further, and so maintain the new higher rate. To lower the fissioning rate, the control rods are pushed deeper into the fuel to absorb more of the neutrons. When fissioning has dropped to the lower rate, the rods are pulled out to their former position.

Figure 21-1 shows a schematic layout of a nuclear reactor which has a fuel core, moderator, shielding, reflector, control rods, and coolant passages. These elements may be arranged in a wide variety of ways in actual reactors.

The *reactor core,* made up of the fuel rods or assemblies, may contain natural uranium metal, enriched uranium, plutonium, or U-233. The fuel rods may be diluted with nonfissionable material to get better reaction control or to minimize damage from fission-product poisoning. The fuel rods may be machined or rolled into a variety of shapes such as tubes, sheets, rods, balls, or powders. They must be clad with a corrosion-resisting metal, such as aluminum, stainless steel, or zirconium. The fuel may also be uranium oxide pellets held in a container made of the same metals just mentioned for cladding. Fissioning makes the fuel metals hot and produces temperatures of the order of 1100 F.

The *moderator* in a reactor core consists of a material, such as graphite, packed round the fuel to slow the fast neutrons to a thermal speed. In some reactors the moderator may be mixed with the fuel. The moderator may be pure graphite, heavy water, or light (normal) water. Beryllium may also be used for compactness. Fast neutrons with an energy level of about 1 million electron volts (1 mev) travel a few inches at speeds of about 8,700 mps. Moderators reduce this speed to a thermal level of about 2 mps in a very small fraction of a second.

Thermal shielding must surround the entire reactor core to absorb some of the radiations (beta particles, escaping neutrons, and gamma rays) produced by the fissioning. The shield, usually made of iron, partly absorbs these energy forms and becomes heated. This prevents the adjacent wall of the reactor vessel from becoming locally heated and warped. The coolant flows over the thermal shield to carry off the generated heat.

The *reflector* usually completely surrounds the reactor core within the thermal shielding. It bounces back most of the neutrons that escape from the fuel core. In Fig. 21-1 the outer edges of the moderator also function as a reflector. Neutrons colliding with the atoms of the reflector heat it, so that it also must be cooled.

The *reactor vessel,* or tank, completely encloses the reactor core with its reflector and thermal shield. It provides the passage for directing the coolant flow through and around the reactor core. The vessel may have to withstand pressures of only a few hundred pounds per square inch or pressures as high as 3,000 psi. The reactor core sits near the vessel bottom to give room for maneuvering the fuel-handling gear above the core. Fuel assemblies (rods) enter and leave the vessel through an access hole at the top. In some water-cooled reactors these fuel assemblies can be manipulated manually through the access door by long tongs. About 15 ft of water must be maintained as a shield between the radioactive fuel and the operators handling it. There are one or more inlet nozzles at the tank bottom to admit coolant flow and one or more nozzles at the upper edge to discharge coolant flow.

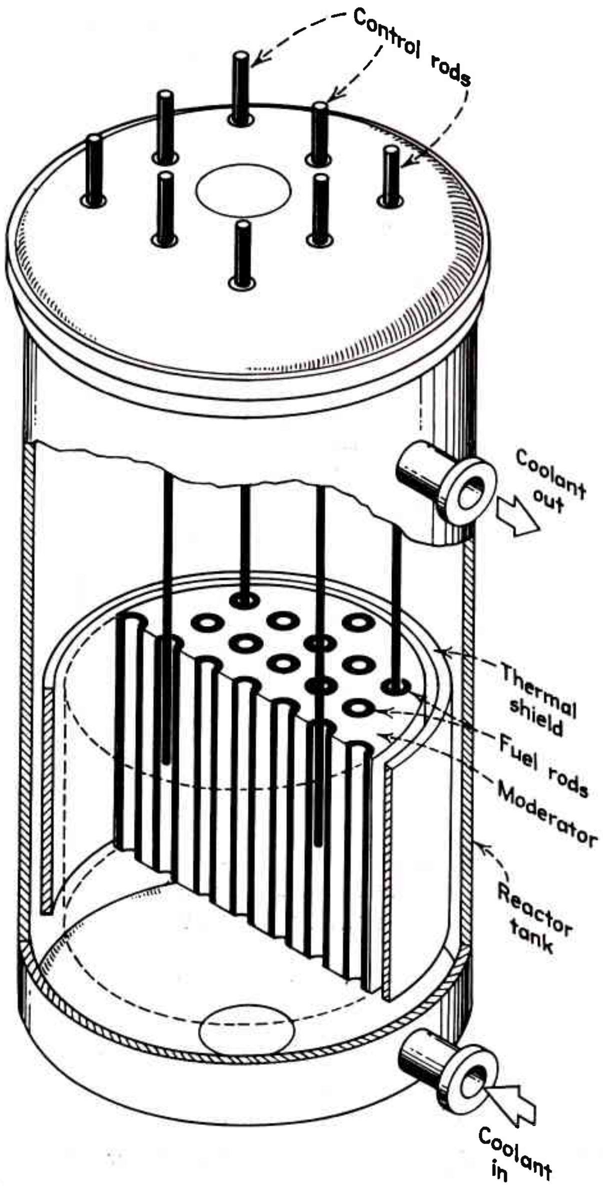


FIG. 21-1. Schematic arrangement of power reactor shows uranium fuel rods held in moderator surrounded by thermal shielding and supported in a reactor tank or vessel. Coolant flowing upward through fuel-rod tubes carries off heat developed by fissioning U-235 atoms. Control rods hold fissioning at proper level by absorbing excess neutrons.

Coolant flow performs two duties: (1) it carries away the useful heat produced by fission to power-generating apparatus such as a steam boiler, and (2) it keeps the reactor fuel assemblies at a safe temperature to prevent melting and destruction. Heat must be removed continuously in proportion to the fissioning rate. Coolant fluids comprise a variety of types: air, carbon dioxide, hydrogen, helium, light water, heavy water, fused salts, organic fluids, sodium, and sodium-potassium and lead-bismuth alloys.

The *control rods* of the reactor absorb the excess neutrons released by fissioning. The rods, usually of boron, absorb neutrons instantly and can be moved in and out of holes in the reactor-core assembly. The rods may be cylindrical or in sheet form. Control rods are of two types, *safety* and *shim*. Safety rods are completely drawn out of the core before controlling the fissioning by inching the shim rods out and in. If the reactor should surge accidentally because the shim rods stick, an automatic trip system rapidly forces in the safety rods to halt the chain reaction. This is called "scramming" the reactor.

The *biological shield* may completely encase the reactor to contain and attenuate the lethal radiations produced by fissioning. Lead, iron, or concrete may be used for shielding. Concrete thicknesses must be of the order of several feet but can be reduced by adding iron oxide to the cement. In some designs the biological shield acts as the walls of a pool of water. Water covering the top of the reactor to a sufficient depth acts as the cover shield.

21-4. Reactor Operation. Changing fuel in a reactor involves special methods. Some of the fission fragments acquire high-intensity radioactivity with a half-life of scores of years. After a certain proportion of the fuel has fissioned, the accumulating fission products absorb the fission neutrons and bring the chain reaction to a halt. These spent fuel assemblies must be replaced with fresh fuel. The radioactivity makes it essential to perform this operation remotely to protect the operators.

Experiment at this time shows that 1 ton of natural or slightly enriched uranium can produce 10,000 megawatt-days of heat energy; this equals 240,000,000 kwhr. On the average, when 1 lb of U-235 fissions, 0.00091 lb of its mass converts to energy. This energy then equals

$$11.3 \times 10^9 \times 0.00091 = 10.3 \times 10^6 \text{ kwhr per lb of U-235}$$

Then at 240×10^6 kwhr per ton only $(240 \times 10^6)/(10.3 \times 10^6) = 23.3$ lb of uranium per ton fissions. In other words, this is a burn-up of only $100 \times 23.3/2{,}000 = 1.2$ per cent of total fissionable and fertile material.

A 1 per cent burn-up seems wasteful of uranium, but this will probably be increased over the years. On the other hand, highly enriched uranium can be worked to a burn-up of about 40 per cent according to some predictions; but the basic fuel is much more expensive. If the fuel assembly is in solid form, fissioning deforms it by warping and hardening. This must not proceed to the point that will prevent removing the assembly from the reactor core. Neutron displacement in the core also weakens the cladding and supporting members.

Another factor contributing to low burn-up is the accumulation of fission fragments as fissioning proceeds. Unfortunately they absorb neutrons, and when enough fission products are formed, so many neutrons are absorbed that the chain reaction cannot be supported.

The burned-up radioactive fuel assembly containing 99 per cent of uranium must be removed from the reactor core to "cooling storage" in a deep pool of water where the shorter-lived radiation dissipates during several weeks. During this time the U-239 transmutes to Pu-239. At a processing plant the spent fuel slugs are dissolved in acid to separate the U-235, U-238, and Pu-239 from the fission products. This involves some 30 chemical reactions and many operations.

Many of the radioactive fission products will find use in industry and medicine, where penetrating radioactivity has proved highly valuable. But, at this time, safe disposal of the remainder as a waste poses a costly problem. The fissionable and fertile materials may be used to recharge a reactor for more power production. But they must be processed and machined by remote methods because of their radioactivity.

All this highly complicated processing of fresh and salvaged fuel makes the cost quite high. Since we get such a high energy yield per pound, we can afford to pay much more per pound than for conventional fuels. But if ways are not found to reduce this cost, nuclear plants will be used only where conventional fuels are unavailable or very costly.

With these major facts of nuclear heat production in mind, let us study some practical reactor layouts, first in schematic form and then in plant form.

21-5. Pressurized-water Reactor. Figure 21-2 shows the arrangement of a pressurized-water reactor (PWR) that uses either heavy or light water as both moderator and coolant. The water completely fills the reactor vessel, heat-exchanger tubes of the boiler, and connecting pipelines. The pressurizing tank keeps the water at more than 1,200 psig so that it will not boil.

Electric heating coils in the pressurizer boil off some of the water to form steam that collects in the dome. As more steam is forced into the dome by boiling, its pressure rises and so pressurizes the entire coolant circuit. To reduce pressure, cooling coils or spray water condenses some of the steam.

The pump circulates the coolant into the bottom of the reactor vessel, up through the core passages and over the reflector and thermal shield, out through the outlet nozzle into the heat-exchanger section of the steam boiler, then out to the pump suction to complete the circuit. The pump only circulates the coolant (does not pressurize it) by generating a pressure rise just enough to overcome the flow resistance of the circuit.

As the coolant flows upward through the core, it is heated by the fissioning fuel elements. Some local steam bubbles may form at hotter spots of the fuel elements, but they quickly condense in giving up their heat to the surrounding water coolant. The hot coolant gives up its energy by heat transfer to the feedwater returned from the turbine condenser, in the heat-exchange section of the steam boiler. The cooled coolant then returns to the reactor vessel for reheating.

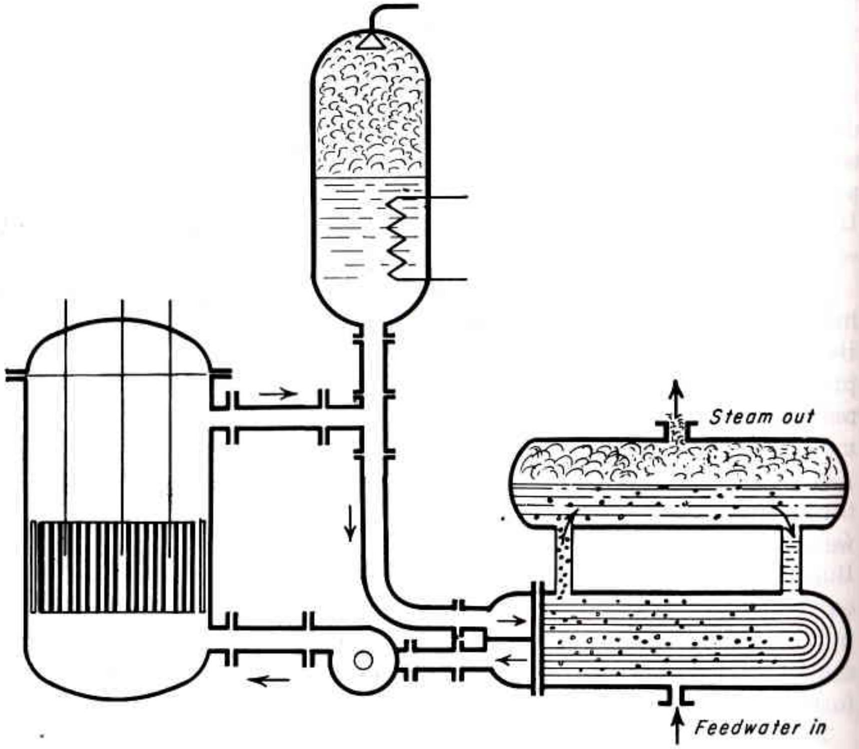


FIG. 21-2. Pressurized-water reactor has pump circulating water between reactor vessel and boiler heat exchanger. Water acts as coolant and moderator. Pressurizing tank maintains water pressure.

The coolant becomes radioactive in passing through the reactor core, so that the entire coolant circuit, including the steam generator, must be shielded to protect operating personnel. The radioactive coolant does not have the ability to make the steam in the boiler radioactive.

With the pressurized coolant at about 2,000 psig the highest-pressure

steam is limited to about 600 psig leaving the boiler. This reactor arrangement can produce only saturated steam. The steam, however, can be superheated by separate firing after leaving the boiler.

21-6. Boiling-water Reactor. Figure 21-3 shows the boiling-water reactor (BWR), the simplest form of nuclear reactor. Feedwater returning from the turbine enters the bottom of the reactor vessel to join water in the reactor circulating upward through the core and downward over the thermal shield around the core periphery.

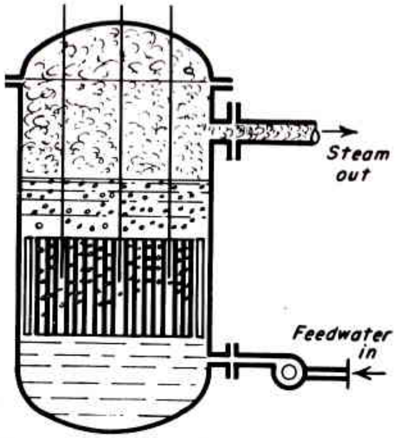


FIG. 21-3. In boiling-water reactor coolant water, also acting as moderator, circulates upward through fuel core and downward over thermal shield. Steam bubbles break through water surface—steam collecting in upper vessel leaves for turbine. Feedwater replaces steam generated.

Steam bubbles form on the surfaces of the fuel elements and are carried upward in the circulating water to break through in the steam-release spacea bove. The steam then leaves to enter the turbine. The water acts as both coolant and moderator.

Steam pressure in this reactor need be only at the working pressure used by the turbine. The reactor vessel can be much lighter than that for a PWR. This reactor does not need a pressurizer, steam generator, circulating pump, or connecting piping. The steam will be hotter because of eliminating temperature drops in a separate steam generator and because it can be produced at a higher pressure.

Steam leaving this reactor will be radioactive, but at a low level and with a half-life on the order of 15 min. The turbine and its piping need light shielding to protect personnel.

21-7. Sodium-Graphite Reactor. Figure 21-4 shows the circuit of the sodium-graphite reactor (SGR), which has two coolant loops. The fuel elements in the core are held in a graphite moderator. Sodium (Na) completely fills the reactor vessel, intermediate heat exchanger (IHX), and connecting piping. The pump circulates the liquid sodium through this primary loop to be heated in the reactor core and cooled in the IHX.

Sodium boils at 1620 F under atmospheric pressure and freezes at 208 F. The entire system must be traced by an electric heating system so that the sodium can be melted prior to circulation by the pump. But sodium has a high boiling temperature, and it need be only moderately pressurized, on the order of 100 psig—this minimizes the cost of the reactor vessel. High sodium temperatures permit high outlet reactor temperatures and maximum steam temperatures through superheating. Since sodium reacts violently with air and water, any system handling it must be leak-tight.

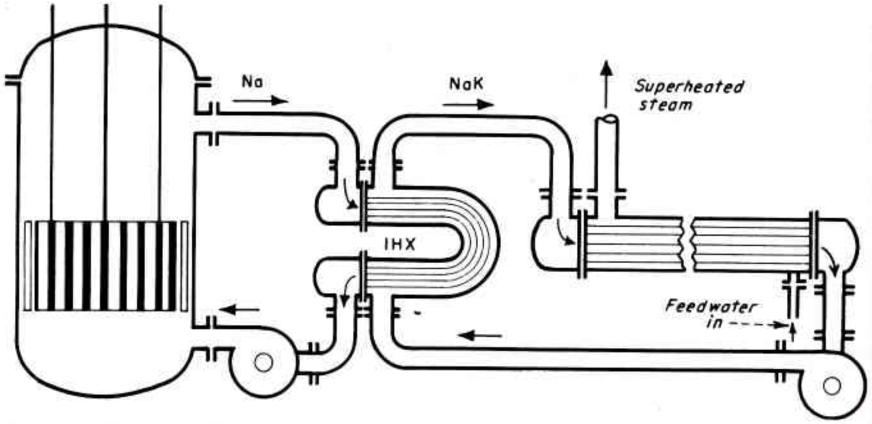


Fig. 21-4. Sodium-graphite reactor uses two liquid-metal coolant circuits, connected through intermediate heat exchanger, IHX, and a once-through steam generator producing superheated steam.

In the IHX the primary sodium loop gives up its heat to an alloyed liquid metal, sodium-potassium (NaK). A pump circulates this between the IHX and the tubes of the steam generator. In the once-through boiler the feedwater boils to steam and then becomes superheated before leaving for the turbine.

Because sodium becomes intensely radioactive in the reactor, the intermediate loop is used to guard against transferring this radioactivity to the steam. This also prevents spreading radioactive vapors in an accident if a leak occurred in the boiler mixing radioactive sodium and water.

In charging or draining one of these loops the liquid metal must be handled under the cover of an inert gas like helium to prevent contact with air.

21-8. Fast-breeder Reactor. Figure 21-5 shows the schematic layout of the fast-breeder reactor vessel. The reactor core has no moderator, uses highly enriched fuel, and is liquid-metal-cooled. This reactor uses two liquid-metal coolant circuits like the SGR.

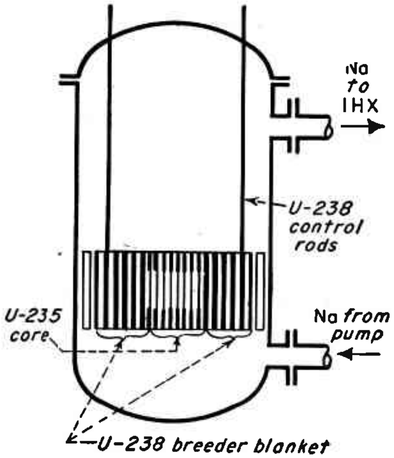


Fig. 21-5. Fast-breeder reactor has enriched core surrounded by depleted uranium blanket. Liquid-metal coolant removes heat from both.

The compact core of U-235 fissions by absorbing the high-speed neutrons released. The central U-235 core is completely surrounded by U-238, which absorbs the excess neutrons and so converts to plutonium. Part of the plutonium participates in the chain reaction and fissions to produce heat. After the central core works to maximum burn-up, the U-238 blanket elements are processed to remove the plutonium, which can be used to fuel another or the same reactor.

The fuel elements making up the U-235 core consist of a central portion of U-235 metal between equal lengths of U-238 metal. The fuel elements inserted in the periphery section of the core contain only the fertile U-238. This arrangement completely surrounds the fuel core with fertile material. Uranium 238 is also used as control-rod material to control the chain reaction—insertion absorbs more neutrons, withdrawal absorbs fewer neutrons.

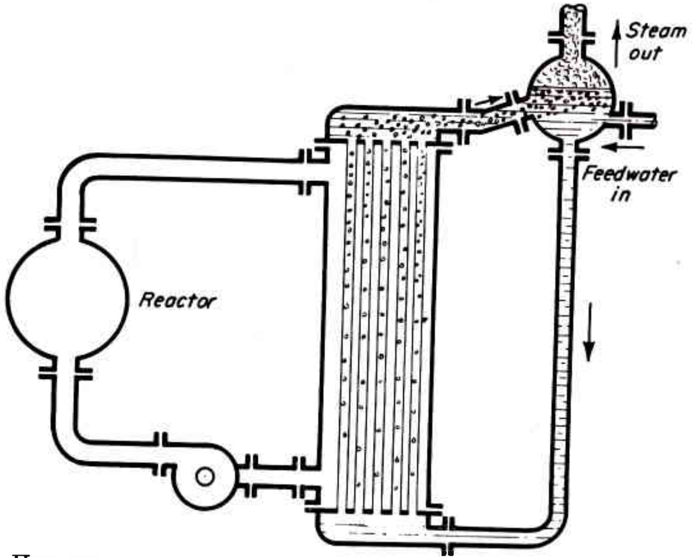


Fig. 21-6. Homogeneous reactor has pump circulating uranium-bearing fluid between reactor and boiler. Critical mass of uranium collects in reactor to heat the fuel-fluid mixture.

21-9. Homogeneous Reactor. Figure 21-6 shows a reactor using a liquid that carries U-235 in a finely divided form, such as: (1) a sulfate of uranium dissolved in light or heavy water, (2) an oxide of uranium carried as a slurry in water, or (3) a solution of uranium in liquid bismuth. The last would have a graphite arrangement in the reactor for a moderator.

The pump circulates the uranium-bearing liquid between reactor vessel and heat exchanger of the steam generator. The reactor vessel just holds a critical mass of uranium that continually changes because of the constant flow. Fissioning U-235 heats the solution in the reactor, but as any part of the solution flows out of the reactor, its mass is below the critical size to support a chain reaction and fissioning stops.

The heated solution enters the boiler to give up its energy to the water circulating through the vertical heat-exchanger section. The pump removes the cooled solution and forces it back into the reactor. The boiler acts like a conventional natural-circulation boiler with a boiler drum to receive incoming feedwater from the turbine and to release steam bubbles from the circulating boiler water.

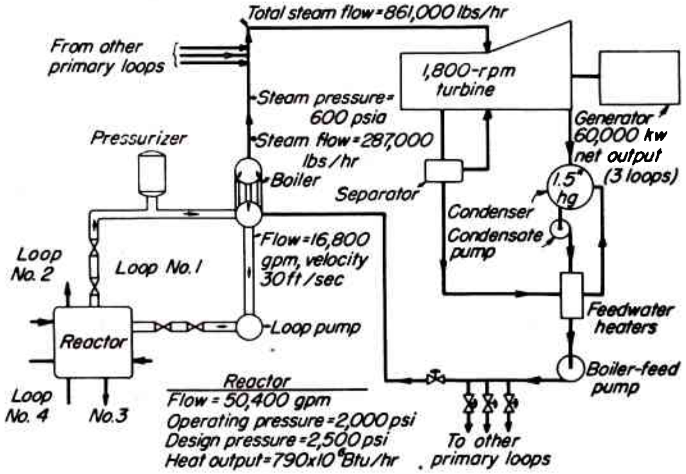

Total steam flow = 861,000 lbs/hr
From other primary loops
1,800-rpm turbine
Steam pressure = 600 psia
Steam flow = 287,000 lbs/hr
Pressurizer
Boiler
Separator
Generator 60,000 kw net output (3 loops)
1.5" hg
Condenser
Condensate pump
Loop No. 2
Loop No. 1
Flow = 16,800 gpm, velocity 30 ft/sec
Loop pump
Feedwater heaters
Reactor
Boiler-feed pump
Reactor
Flow = 50,400 gpm
Operating pressure = 2,000 psi
Design pressure = 2,500 psi
Heat output = 790x10⁶ Btu/hr
Loop No. 4
No. 3
To other primary loops

FIG. 21-7. Shippingport Plant has one reactor and four coolant loops to produce steam for turbine. Separator receives all of partly expanded steam from turbine to remove moisture before returning steam to low-pressure stages. (*Courtesy of Westinghouse Electric Corporation.*)

Cycles using water as the uranium carrier must be pressurized to avoid boiling in the fuel-circulating loop. A liquid-metal–fuel solution need not be pressurized but must have a graphite moderator in the reactor. In this arrangement the fuel can be continuously processed to remove fission products and inject fresh fuel.

21-10. Gas-cooled Reactor. The circuit for a gas-cooled reactor is similar to that shown in Fig. 21-2, with these differences. The reactor core would include a graphite moderator. The coolant would be a gas and the pump a blower or gas compressor. There would be no need for a pressurizer. The vessels and piping would be considerably larger because of the lighter density of the coolant. The tubes of the boiler would have extended surfaces such as fins to improve heat-transfer rates. Gas pressures would be less than 100 psig, so that vessel wall thickness would not be a problem. The gas could be air, hydrogen, helium, or carbon dioxide.

21-11. Shippingport Plant. This was the first large-capacity nuclear plant to start operating in the United States. The plant has a 100-mw outdoor turbine supplied with 600-psig saturated steam from four steam generators. All four boilers get their heat from one PWR through coolant loops running at 2,000 psig (Fig. 21-7).

Reactor, pressurizer, coolant pumps, and steam generators stand in four large containment tanks. These tanks can hold all the vapor and gases that might be released by a major accident to any of the equipment. The tanks are a precaution against spreading radioactive gas that might be produced by an accident.

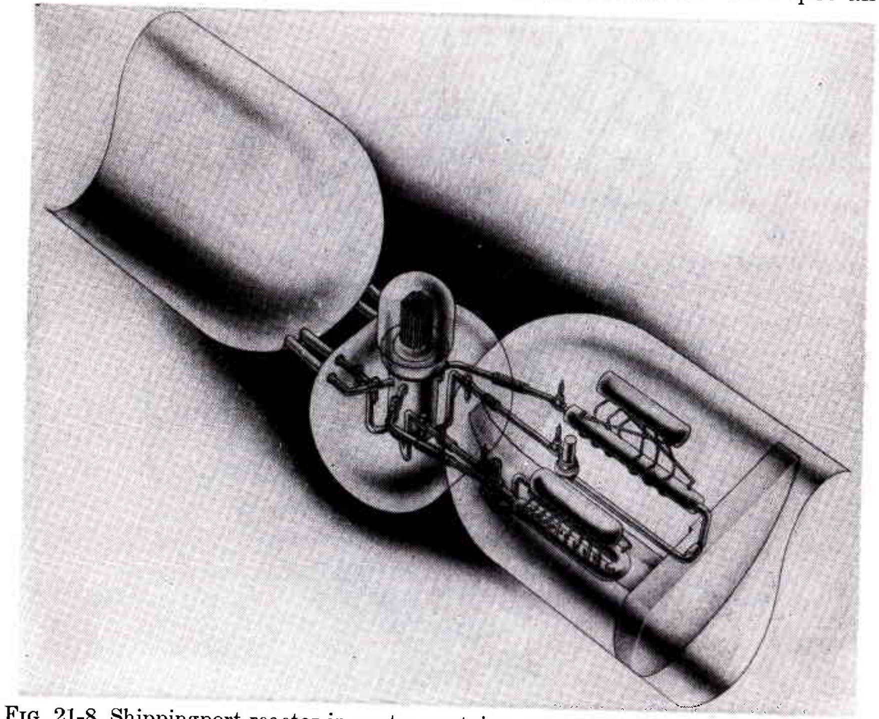
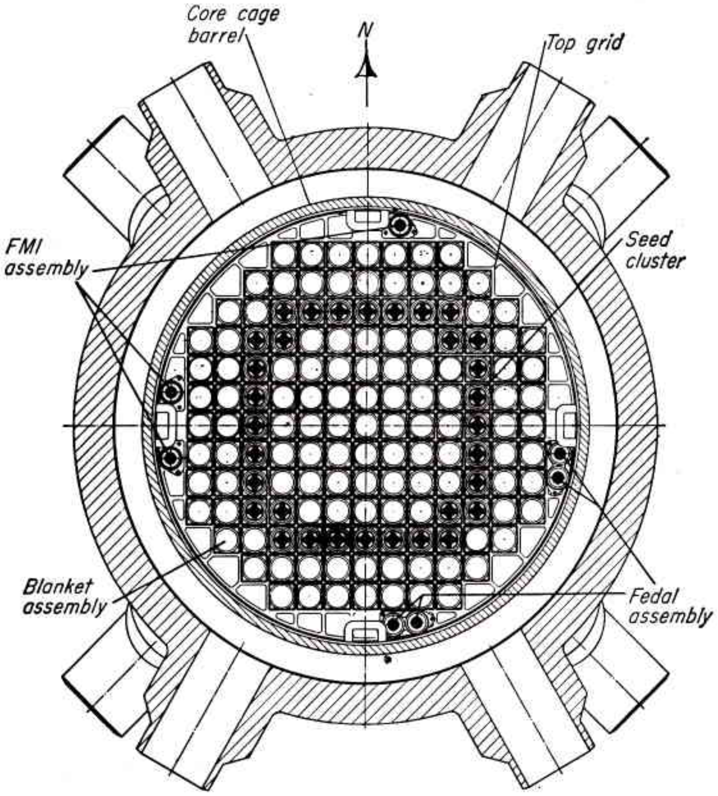


FIG. 21-8. Shippingport reactor in center containment vessel feeds four coolant loops, arranged two each in adjacent containment tanks. (*Courtesy of Westinghouse Electric Corporation.*)

Figure 21-8 shows three of the containment tanks, with the reactor and two of the boilers indicated. Concrete walls separate the two boilers in a common containment tank. This protects maintenance men while working on one coolant-circuit pump or boiler from radiation that might emanate from the operating circuit equipment. Other auxiliaries and the pressurizer stand in a fourth tank not shown. All containment tanks are connected together by piping so that the total volume will be available for gases released in any tank.

Figures 21-9 and 21-10 show the reactor-core and vessel assembly. Two kinds of fuel elements, (1) seed assemblies and (2) blanket assemblies, and cross-shaped control rods make up the reactor core. The 24 seed assemblies consisting of 95 per cent U-235, alloyed with zirconium, are arranged in a square. The cross-shaped control rods move up and down within these assemblies.



Pressure vessel–cross section above core

FIG. 21-9. Shippingport reactor core has 24 U-235 seed assemblies with cross-shaped control-rod inserts and 93 UO_2 blanket assemblies. (*Courtesy of Westinghouse Electric Corporation.*)

Four subassemblies welded together with spacers make up each seed assembly. The subassemblies look like the element at the right of Fig. 21-11. The 2,000-psig coolant passes through the channels and over all surfaces of the subassembly. All surfaces of the U-235–zirconium alloy are clad with Zircaloy-2 to guard against corrosion.

Blanket assemblies fill the space inside and outside the square formed by the seed assemblies. Zircaloy-2 tubes filled with UO_2 pellets make up the blanket assemblies. Each blanket is a stack of seven bundles of 100 tubes. Figure 21-11, left, shows a bundle held together by end plates. The 2,000-psig water flows up through the holes in the end plates to cover the UO_2-filled tubes.

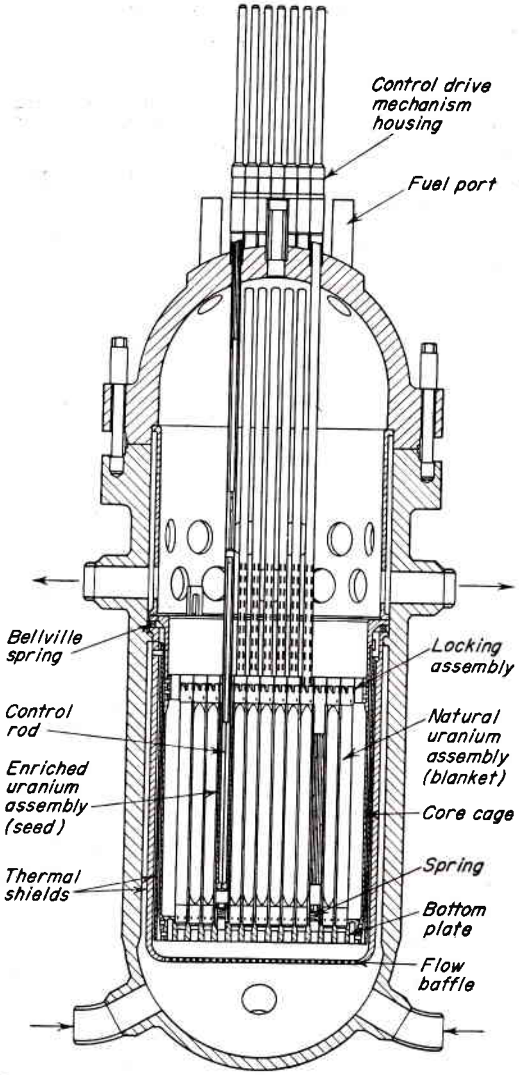


FIG. 21-10. Shippingport reactor vessel has core in bottom half with control rods passing up through the dome. Coolant (2,000-psig water) enters at bottom ports, flows up through core, and leaves through center ports. (*Courtesy of Westinghouse Electric Corporation.*)

All the fuel elements lock into the bottom support plate and the upper grid (Fig. 21-10). The 2,000-psig water enters the bottom of the reactor vessel to flow up through the flow baffle into the fuel elements. About 10 per cent of the water bypasses the core to cool the vessel wall and thermal shield. The water itself acts as a shield in this area. Water enters the core at 508 F and leaves at 542 F.

The reactor vessel has a 9-ft ID and 8½-in. wall of carbon-steel plate and forgings with ¼-in. stainless-steel cladding. The vessel stands 33 ft high with a 6-in.-thick bottom and a 10-in.-thick cover bolted on with forty-two 6-in. studs.

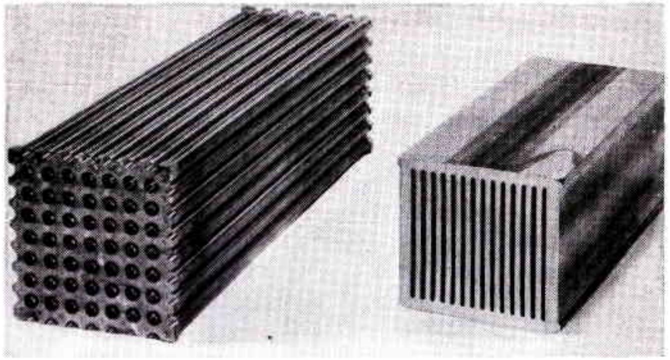


FIG. 21-11. Left, uranium oxide pellets encased in Zircaloy tubing held in end plates are cooled by water flowing over outer surface. Right, Zircaloy-clad U-235 element has 14 coolant channels. (*Courtesy of Westinghouse Electric Corporation.*)

Control-rod drives and fuel ports for removing fuel assemblies pierce the top cover. Fuel assemblies can be removed and inserted individually through the fuel ports. By lifting off the top cover the whole core can be removed from the vessel. The containment tank surrounding the vessel holds a 3-ft thickness of shielding water. In turn the concrete shielding around the reactor containment tank holds additional shielding water, which covers the reactor vessel and cover to about 25-ft depth. The entire assembly stands below ground level.

Isolation valves next to the reactor inlet and outlet for each of the four coolant loops make it possible to isolate a loop for repairs without shutting down the reactor. The eight valves are remotely and hydraulically controlled.

Pumps handling the radioactive coolant must have zero leakage. The canned-motor pump (Fig. 21-12) meets this specification. Both stator and rotor of the driving motor are hermetically sealed in cans or sheathing. The water moved by the pump also lubricates the bearings for both motor and pump. Bearings must have a large enough surface to compensate for the poorer lubricating qualities of the water.

Water completely fills the air gap in the motor and any other spaces. The sealing sheaths add to the motor air-gap loss. Cooling coils surround the exterior of the stator winding to prevent their overheating. The electrical insulation must resist nuclear radiations given off by the water.

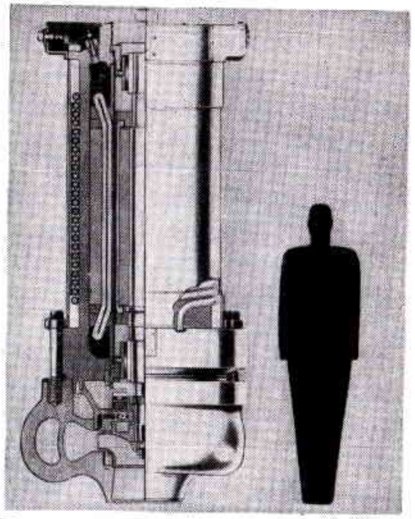


FIG. 21-12. Canned-motor pump for 2,500-psig design pressure handles 16,800 gpm of water coolant. (*Courtesy of Westinghouse Electric Corporation.*)

A 1,200-kw two-speed 2,300-volt three-phase motor drives the single-stage 15,000-gpm centrifugal pump made of stainless steel. The motor, weighing 20,000 lb, mounts in the cast stainless-steel volute with a seal-welded joint.

21-12. Steam Generators. Steam generators of two designs (Figs. 21-8 and 21-13) work with the Shippingport reactor. Both have a heat-exchanger section and a steam drum. The high-pressure coolant flows through several hundred small-diameter stainless-steel tubes in the exchanger section. The coolant cools from 542 to 508 F to form steam bubbles in the boiler water surrounding the tubes. A mixture of steam bubbles and boiler water passes through the several risers into the drum above. Steam bubbles break away from the water in the drum and pass

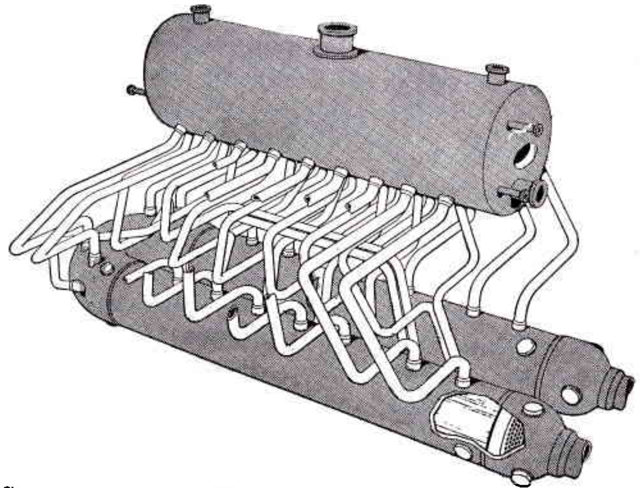


FIG. 21-13. Steam generator has U-shaped heat exchanger at bottom and steam drum above to receive feedwater and separate water and steam.

through separators to remove any entrained water. The 600-psig steam leaves almost dry for the turbine. The boiler water returns to the exchanger through the downcomers.

Boilers produce 263 million Btu per hr in the steam. At no load the steam pressure rises to 885 psia and drops to 600 psia at full load. The 2,000-psia coolant flows through the exchangers at the rate of 6.5 million lb per hr. Two of the boilers are entirely of stainless steel with a straight-tube and fixed-tube-sheet design; they are 36 ft long and 43 in. in diameter.

The other two boilers (Fig. 21-13) have return-bend U-tube exchanger sections with an over-all length of 28 ft and a 39-in. diameter.

21-13. Pressurizer. The pressurizer for Shippingport is a vertical tank 18 ft high with a 5-ft ID and 300-cu ft volume. About 150 cu ft acts as steam-dome volume at maximum water level. Normal surges will be as high as ±10 cu ft, corresponding to 1,850- and 2,185-psia pressure limits. The lower pressure must be above a level fixed by core hot-spot temperature, and the higher pressure should be below relief-valve setting.

The 500-kw electric heaters in the tank supply the energy for initially raising coolant-system pressure. The system must be filled to the proper level in the pressurizing tank with highly purified water. Closing the system and starting the heaters heats the water in the tank till it boils. Boiling at constant volume gradually raises steam and water pressure to the operating level at 2,000 psig.

21-14. Argonne Experimental Boiling-water Reactor. Figure 21-14 shows the experimental BWR at the Argonne Laboratory in Lemont, Ill. All the equipment except the central control board stands inside the containment vessel. The vessel, 123 ft high and 80 ft in diameter, is made of welded steel plate lined with concrete. It will hold gases and steam that might be released in any accident.

A 6,250-kw-capability turbine generator stands on the operating floor with its main condenser directly underneath. Next to the condenser stands the steam separator for removing moisture in the main steam leaving the BWR, which stands just to the right of center.

The reactor vessel is 7 ft in diameter, 25 ft high, and 2½ in. thick and is forged of boiler steel with an inside layer of stainless steel. Stainless-steel wool insulates the vessel. Three inches of water-cooled lead, and several feet of special concrete, protect the surroundings from radiation.

In the bottom half of the vessel stands the reactor core, containing 12,500 lb of slightly enriched uranium alloyed with niobium and zirconium and clad with zirconium. The fuel elements are plates about ¼ in. thick, 3¾ in. wide and 4 ft long.

The control rods of hafnium and boron-bearing stainless steel slide into the core from the bottom. The control-rod mechanism stands underneath the reactor vessel at the bottom of the containment vessel. The reactor vessel holds 3,000 gal of water and was designed to generate 60,600 lb of steam at 600 psig (489 F), or a total of 20,000 kw of heat.

The reactor first went critical on Dec. 21, 1956; seven days later it produced its rated output. After a year of experimentation the reactor output was raised to 40,000 kw of heat on Dec. 2, 1957. Then on Mar. 20, 1958, the reactor produced 61,700 kw of heat with the original core. The designers estimate that with minor modifications of the core the reactor can produce as much as 100,000 kw of heat.

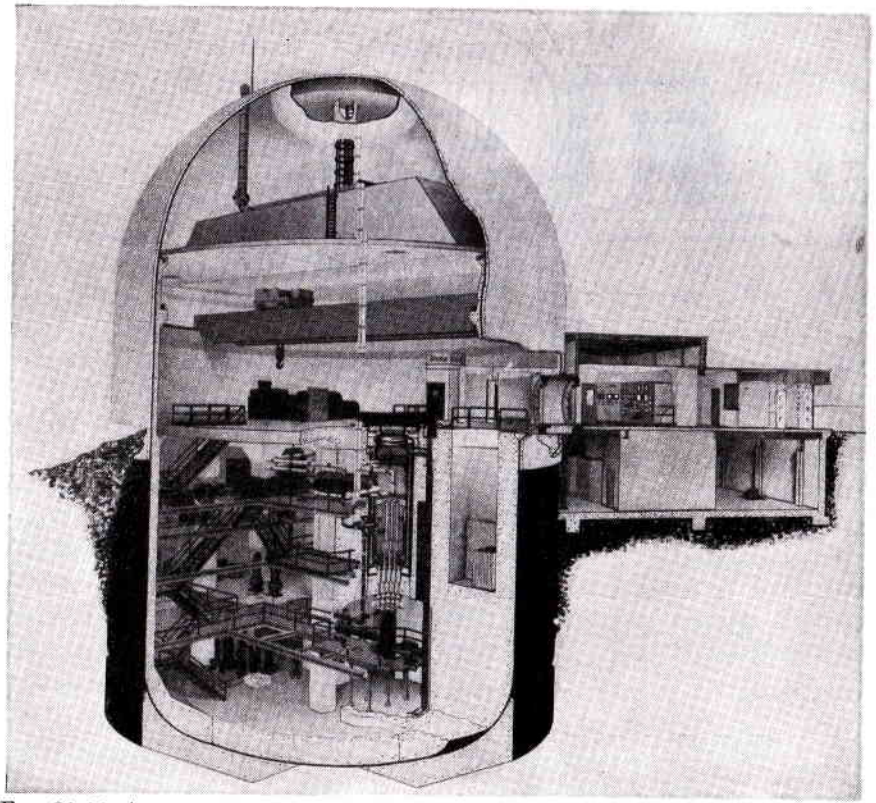


FIG. 21-14. Argonne experimental boiling-water reactor plant has 6,250-kw turbine supplied-600-psig steam from reactor. Containment tank completely encloses plant to prevent escape of radioactive vapor that might be released in an accident. (*Courtesy of Argonne National Laboratory.*)

A rail-mounted lead coffin, not shown in Fig. 21-14, transports radioactive fuel rods from the reactor to the water-filled decay storage pit at the right. Prior to the removal operation, the reactor is completely filled, up to the operating floor, with cold treated water, the cover having been removed. The water gives the operators biological shielding and permits examination of the reactor interior. Tongs within the lead coffin reach into the reactor and pull up spent fuel assemblies into the coffin. Fresh unirradiated fuel assemblies can be inserted manually into the core with the aid of long tongs.

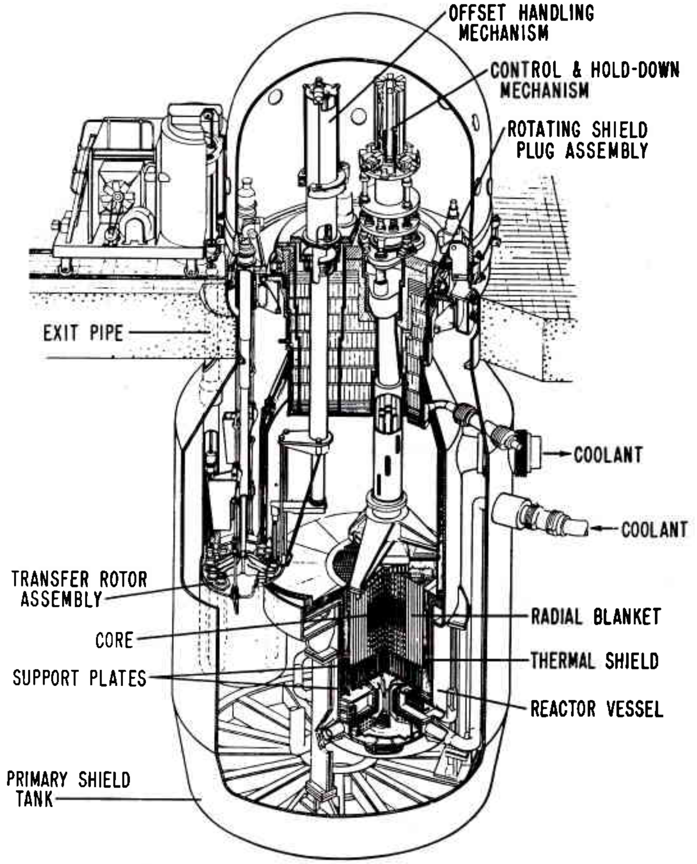


Fig. 21-15. Enrico Fermi Plant reactor has enriched core surrounded by blanket of depleted uranium. Reactor vessel is completely filled with liquid sodium, which acts as coolant. Rotating shield plug at top of vessel holds equipment to transfer fuel assemblies in core.

21-15. Enrico Fermi Plant. This plant uses a fast-breeder reactor discussed in Sec. 21-8 and shown in Fig. 21-5. The reactor has three coolant loops each with an IHX, circulating pumps, and steam generator. The secondary coolant is also sodium.

Figure 21-15 shows the reactor of the plant together with the fuel-handling equipment. The reactor core produces most of the heat and about 25 per cent of the Pu-239. The core consists of an assembly of alloy pins of uranium enriched to 27 per cent U-235 and alloyed with 10 per cent of molybdenum. Alloy rods containing depleted uranium completely surrounds the core, to form the breeder blanket. The blanket produces about 75 per cent of the Pu-239. For every pound of U-235 fissioned to produce heat this reactor converts 1.2 lb of U-238 to Pu-239.

A hold-down plate on top of the core keeps the fuel elements aligned and prevents them from floating upward with the sodium coolant passing through. The plate also keeps the control-rod drives aligned. The 10 control rods, made of boron, regulate the core power level. Two of these are shim rods; the other 8 are safety rods.

The top of the reactor vessel is closed by a rotating shield plug. To remove an element from the core, the control rods are locked in and the drives disconnected remotely. The hold-down plate is lifted and the shield plug rotated so that the offset handling mechanism can latch onto the fuel element to be removed. This mechanism then lifts out the fuel element and deposits it in the transfer rotor. All this must be done remotely since the reactor vessel is filled with radioactive sodium. A transfer cask car removes the fuel element from the transfer rotor. New elements are inserted by the reverse method. The transfer rotor container is an integral part of the reactor vessel.

The reactor vessel and its auxiliary handling devices are completely enclosed in a primary shield tank to guard against distributing radioactive gases in an accident. This assembly and primary-loop pumps and IHXs in turn are enclosed in a gastight building or containment tank.

Sodium from each of the loops enters the reactor vessel at the bottom to flow up through the core. The sodium then flows through the outlet to the IHX. From the bottom of the IHX it passes to the circulating pump suction to repeat the cycle.

The principal characteristics of the plant are:

Electrical capacity	100 mw
Reactor heat output	300 mw
U-235 critical mass	1,070 lb
Fuel alloy	90% U-10% Mo
Uranium enrichment	27%
Core diameter	30.5 in.
Core length	30.5 in.
Core avg heat flux	665,000 Btu per hr per sq ft
Core power density	78.5 × 10⁶ Btu per hr per cu ft
Sodium velocity	32.5 ft per sec
Net thermal efficiency	30%
Breeding ratio	1.2
U-235 burned per year	189 lb
Pu-239 produced per year	224 lb
Primary sodium temperatures:	
Leaving reactor	800 F
Entering reactor	550 F
Sodium flow	13,200,000 lb per hr
Secondary sodium temperatures:	
Entering steam generator	750 F
Leaving steam generator	500 F
Steam pressure	600 psia
Steam temperature	740 F

PART III

INTERNAL-COMBUSTION PLANTS

SECCION CENTRAL HIDRAULICA
U.T.E.
* RINCON DEL BONETE *

CHAPTER 22

INTERNAL-COMBUSTION ENGINES

22-1. Relative Capacity. Over 90% of the prime-mover capacity in the United States is in the form of internal-combustion (i-c) engines. These include engines in automobiles, trucks, buses, locomotives, tractors, ships, and stationary plants. The mobile applications by far have the major part of the capacity. We shall be concerned only with the engines used for stationary power generation.

While there are about an equal number of both steam central stations and i-c-engine central stations, the latter have only about 2 per cent of the total national capacity. Steam stations supply about 80 per cent of the total, the remainder being hydroelectric plants.

Reciprocating i-c engines of all types use the same basic processes. Either air or a mixture of air and vaporized fuel is compressed in a cylinder by a piston. The compressed air or mixture is ignited by injecting fuel or by an electric spark to raise the pressure sharply. The pressurized combustion gas then works on the piston while it expands. Finally the spent combustion gas exhausts from the cylinder and a fresh charge of fluids repeats the cycle.

22-2. Engine Classification. Engines may be classified in many ways according to their basic cycle and physical components. Thermodynamically they may be classified according to the combustion process: (1) constant volume—Otto cycle; (2) constant pressure—diesel cycle; (3) combination constant volume and constant pressure—dual, or Sabathé, as outlined in Secs. 3-7 and 3-8.

From the design standpoint engines may be classified as follows:

1. Piston strokes per cycle:
    - *a.* Two
    - *b.* Four
2. Fuel used:
    - *a.* Gasoline
    - *b.* Gas
    - *c.* Oil
    - *d.* Gas and oil combined

3. Ignition method:

   a. Spark-ignition (s-i)
   b. Compression-ignition (c-i)
   c. Surface-ignition

4. Fuel admission:

   a. Injection (air blast or mechanical)
   b. Carburetor
   c. Mixing valve
   d. Compression (for gaseous fuels)

5. Air admission:

   a. Simple aspiration (four-stroke cycle)
   b. Scavenging by blower or pump (two-stroke cycle)
   c. Supercharging (four-stroke cycle)

Many firms build i-c engines. Table 22-1 lists the wide variety of engines available today according to their commercial or common name and gives the basic feature of each.

22-3. Engine Cycles. The cycles described in Secs. 3-7 and 3-8 are based on the *air standard* for easy study. While they do show the general processes in the engine and the effect of various factors, they are fictitious from the practical standpoint. Figure 22-1 shows *indicator* and *pressure-time* diagrams for practical s-i and c-i engines. The net area of the indicator card $\int p\,dv$ measures the net work per cycle done on the piston face by the combustion gases. This is

$$ihp = \frac{imep \times L \times A \times n}{33{,}000} \tag{22-1}$$

where ihp = indicated power per cylinder, hp
$imep$ = indicated mean effective pressure, psi
L = piston-stroke length, ft
A = piston area, sq in.
n = number of cycles per min per cylinder
= rpm for 2-stroke cycle
= ½ rpm for 4-stroke cycle

The piston delivers the indicated power to the engine shaft through a mechanism that converts the reciprocating motion to rotary motion. The net power output is

$$bhp = ihp \times \eta_m \tag{22-2}$$

where bhp = brake horsepower or net delivered power, hp
η_m = mechanical efficiency, fraction

TABLE 22-1. COMMERCIAL INTERNAL-COMBUSTION ENGINE TYPES[a]

Common name	Fuel	Ignition method	Compression ratio[b]	Fuel admission	Air-fuel mixture formed
Diesel engine	Fuel oil	Compression	High	High-pressure injection	In cylinder—end of compression
Gas engine	Gas	Spark	Low	Mixing valve	Outside—before compression
Gasoline engine	Gasoline	Spark	Low	Carburetor	Outside—before compression
Injection gas engine	Gas	Spark	Low	Low-pressure injection	In cylinder—during compression
Injection gasoline engine	Gasoline	Spark	Low	Low-pressure injection	Outside cylinder—before compression
Hesselman engine	Fuel oil	Spark	Low	High-pressure injection	In cylinder—before compression
Vaporizing oil engine	Fuel oil	Hot spot	Low	High-pressure injection	In cylinder—end of compression
Gas diesel engine	Gas	Pilot oil	High	High-pressure injection	In cylinder—end of compression
Dual-fuel diesel engine	Gas and fuel oil	Pilot oil	High	High-pressure compression	In cylinder—end of compression
				Induction	Outside—before compression

[a] Adapted from B. G. A. Skrotzki, "Electric Generation—Hydro, Diesel, and Gas-turbine Stations," McGraw-Hill Book Company, Inc., New York, 1956. (*Courtesy of G. C. Boyer.*)

[b] High (usually above 10); low (usually below 10).

The brake horsepower may also be computed from

$$bhp = \frac{L \times A \times n \times N \times \rho_a \times \eta_v \times (F/A) \times HHV \times \eta_{ti} \times \eta_m \times J}{33{,}000} \qquad (22\text{-}3)$$

where n = number of cycles per min per cylinder
N = number of cylinders
ρ_a = density of entering air, lb per cu ft
η_v = volumetric efficiency
F/A = fuel-air ratio
HHV = heating value of fuel, Btu per lb
η_{ti} = indicated thermal efficiency of actual engine, fraction
A = piston area, sq ft
L = piston-stroke length, ft
J = 778 ft-lb per Btu

This equation develops the brake horsepower in terms of the factors directly controlling it. The thermal efficiency reflects the effects of heat losses, valve throttling, and timing.

22-4. Engine Fuels. Internal-combustion engines use both oil and gas fuels, the former being predominant. Spark-ignition engines use gasoline, and compression-ignition engines use the heavier distilled oils. Some slow-speed large diesels burn residual oils, but the medium- and high-speed engines use a distillate or other gas oil. Engines are usually designed to burn specific types of fuel for best performance. The oils may be specified according to these properties:

Cetane number. An index of ignition quality; indicates the ease with which fuel ignites when injected, and speed of combustion.

Distillation test. Indicates whether oil contains any heavy ends of the refining process, which generally burn poorly.

Aniline point. A measure of paraffinicity; used in calculating the diesel index, which measures ignition quality.

Conradson carbon. Indicates extent of components in oil with a tendency to form carbon deposits.

Viscosity. Significant in oil-handling injection-system performance and in lubrication.

Ash. Components of oils believed to cause cylinder wear.

Water and sediment. Cause of irregular operation and engine wear and corrosion.

Sulfur. Considered to be the factor causing acid-type corrosion if in sufficient quantity.

Volatility. An index of ease with which a combustible fuel-air mixture can be prepared.

Pour point. Indicates temperature at which oil flows.

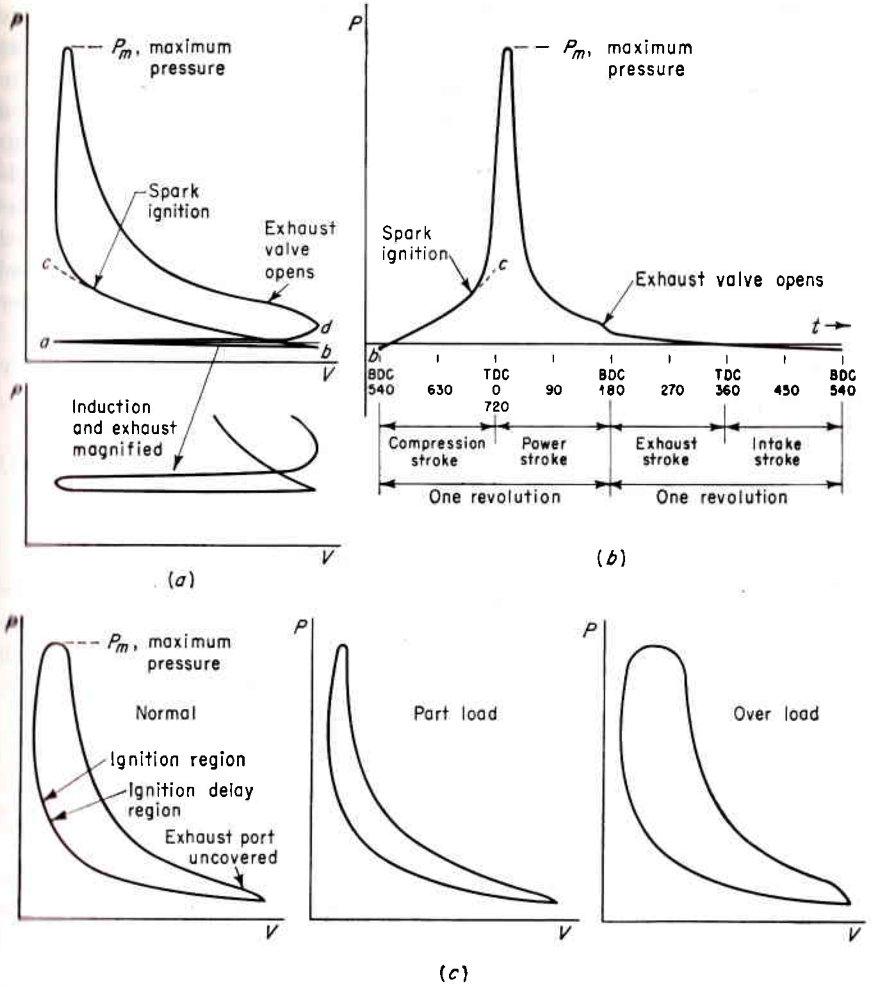


FIG. 22-1. Actual engine cylinder performance. (*a*) Normal full-throttle *P-V* (pressure-volume) diagram for s-i engine; note magnified detail of induction and exhaust for four-stroke cycle. (*b*) *P-t* (pressure-time) diagram comparable to (*a*); note that this type of diagram is obtained by an oscilloscope. (*c*) General nature of c-i engine performance; absence of induction and exhaust strokes indicates a two-stroke cycle. Effects of governing change the diagram area for operation at different loads. This effect is similar for all types of engines.

Flash and fire point. Used mostly in relation to the fire hazards of an oil.

Specific gravity. Expressed in terms of the API scale at 60 F; gives approximate indication of heating value.

Heating value. Measures the thermal energy in fuel.

The *ignition quality* of a fuel plays an important part in smooth operation of a c-i engine. It controls the ease of starting an engine, but, more importantly, the short time available for burning the fuel means that it must ignite and burn quickly. Poor ignition quality will contribute to rough operation. The cetane number to measure ignition quality of a fuel is determined in a standard test engine. Its performance is compared with that of a test fuel compounded of cetane ($C_{16}H_{34}$) and α-methylnaphthalene ($C_{11}H_{10}$) in the test engine. The compression ratio of the engine is varied until the same ignition delay occurs for both the fuel under test and the test fuel. The percentage of cetane in the test fuel having the same ignition delay and compression ratio as the fuel under test is the cetane number of the fuel.

The higher heating value (HHV) of a fuel oil may be obtained by calorimeter tests or calculated by

$$HHV = 17{,}680 + 60 \times {}^\circ API \tag{22-4}$$

where HHV = higher heating value, Btu per lb
$^\circ API$ = gravity, $^\circ API$ at 60 F

Gas fuel for i-c engines may be natural gas, blast-furnace gas or coke-oven gas or other industrial by-product gases, or sewage gas. Dual-fuel engines have been developed that can burn either gas or oil or both so that they can take advantage of fluctuating prices of these fuels. These engines mostly use compression ignition.

Most natural-gas supplies have a heating value of about 1000 Btu; the precise value varies with the gas composition. The heating value is given in Btu per cubic foot at standard conditions of 60 F and 30 in. Hg pressure.

22-5. Compression-ignition Engine Combustion. The c-i engine combustion starts by spontaneous ignition of the fuel and air because of the high temperature developed by the compression of the air. The fuel under pressures as high as 30,000 psia injects as a spray into the pressurized air. The spray has a core of fuel oil surrounded by a turbulent zone of air and fuel particles. The combustion process has four distinct phases: (1) ignition delay, (2) uncontrolled pressure rise, (3) controlled pressure rise, and (4) expansion-stroke combustion.

The ignition delay is the period from the instant when injection starts until the time that the fuel in the cylinder ignites. During this period the injected fuel rises in temperature, mixes with air, atomizes and vaporizes, and starts reacting with the O_2 of the air relatively slowly. Ignition-delay time increases with viscosity of the fuel and is affected by air density, engine speed, shape of combustion space, air turbulence, and other factors. The fuel-air mixture in the cylinder is not uniform; at the end of the delay period the flame appears at random points throughout the mixture. The fuel in the cylinder burns rapidly to increase the pressure rapidly after ignition delay (Fig. 22-2).

In the third phase of combustion the fuel, continuing to enter the burning gases, ignites immediately and burns. Controlling the injection rate in this phase controls the rate of pressure rise. Because of a nonhomogeneous mixture of fuel and air, combustion continues as *afterburning* during the early part of the expansion process. These phases give the c-i engine card (Fig. 22-1), a variable-pressure combustion process.

Spontaneous burning in a c-i engine after the ignition delay may cause *knocking*, which becomes more serious with increasing ignition delay and ignition temperature. A fuel with short ignition delay, high cetane number, and low ignition temperature produces the least knocking.

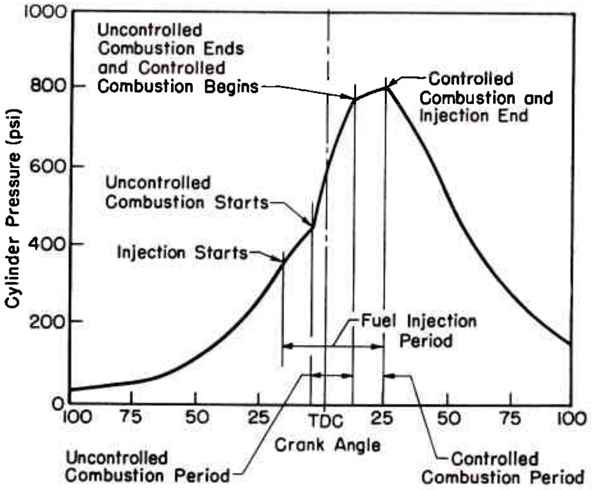


FIG. 22-2. Combustion in a c-i engine cylinder passes through several significant phases.

The diesel, gas-diesel, or dual-fuel engines are of the c-i type. The vaporizing oil engine ignites the fuel-air mixture by maintaining an area of the cylinder end hot enough for ignition.

22-6. Spark-ignition Engine Combustion. The s-i engine uses a spark plug to ignite the fuel-air mixture. The flame advances best through a slightly rich fuel-air mixture since it is diluted by the exhaust gas that remains in the cylinder at the end of the exhaust stroke. The spark creates a flame that advances through the mixture in a spherical pattern.

Flame-front velocity depends on mixture turbulence, incidental cooling, flame-front direction, and other factors. Flames advance at about 30 ft per sec through quiescent mixtures, but this would be suitable only for low-speed engines. Turbulence promotes the higher flame velocities needed by high-speed engines.

The shaping of the combustion chamber and adequate velocity of the mixture entering through the intake valves during the intake stroke produce some of the turbulence needed. Igniting and propagating the flame front requires time. To develop maximum pressure at the start of the expansion stroke, the spark ignites the mixture before the end of the compression stroke (Fig. 22-1*a*). This is called *advancing the spark*. Figure 22-3 shows the effect of engine speed on spark advance.

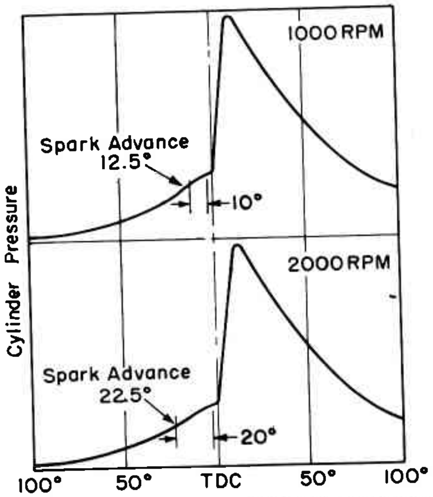
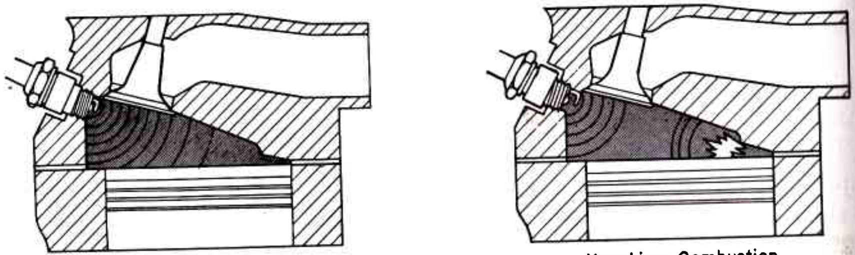


FIG. 22-3. To ensure maximum pressure at the start of the expansion the spark advance must be increased as the engine speed increases.

For normal nondetonating combustion the flame front spreads smoothly throughout the combustion space. It usually slows as it approaches the opposite walls of the combustion chamber.

As the flame front advances through the mixture, it acts like a piston and compresses the unburned part of the mixture. The flame also transfers heat into the unburned mixture to raise its temperature above the compression temperature. This causes spontaneous ignition at one or more points in the unburned mixture and produces local high-pressure regions. This in turn causes rapid pressure rises and pressure waves in the cylinder. These effects produce *knock* in the engine. Knocking produces large impacts on engine parts and possibly vibrates the chamber walls.



Non-Knocking Combustion

Knocking Combustion

FIG. 22-4. Nonknocking combustion results from smooth propagation of the flame front in a cylinder. Knocking combustion is caused by autoignition at several points in a cylinder. (*Courtesy of General Motors Corporation.*)

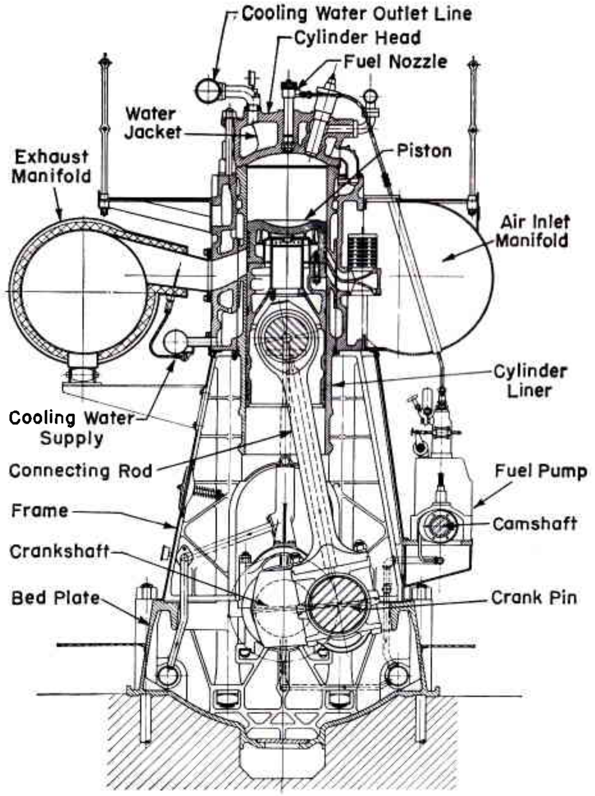


FIG. 22-5. Cylinder and engine cross section of typical two-stroke diesel engine. Note relationship of structural parts mentioned in Table 22-2. (*Courtesy of Nordberg Manufacturing Co.*)

TABLE 22-2. MATERIALS FOR PRINCIPAL ENGINE STRUCTURAL PARTS

Structural member	Cast iron	Meehanite iron	Welded steel	Forged steel	Steel	Aluminum
Bedplate	x		x			
Frame	x		x			
Cylinder head	x					
Cylinder liner	x				x	
Piston[a]	x			x		x
Piston rings	x					
Connecting rod				x		
Crankshaft		x	x	x		
Wrist pin				x		
Valves				x		
Camshaft and cams	x			x	x	

[a] Pistons commonly have cast-iron skirts and forged steel tops.

Figure 22-4 shows the autoignition, or knocking, in a s-i engine. A hot spot at the spark plug or elsewhere in the combustion chamber may cause *preignition* before the normal spark ignition. Detonation at a given compression ratio varies with the gasolines, timing, turbulence, and spark-plug location. Generally, the higher the *octane* rating of a fuel, the higher the compression ratio at which the engine will detonate, or knock.

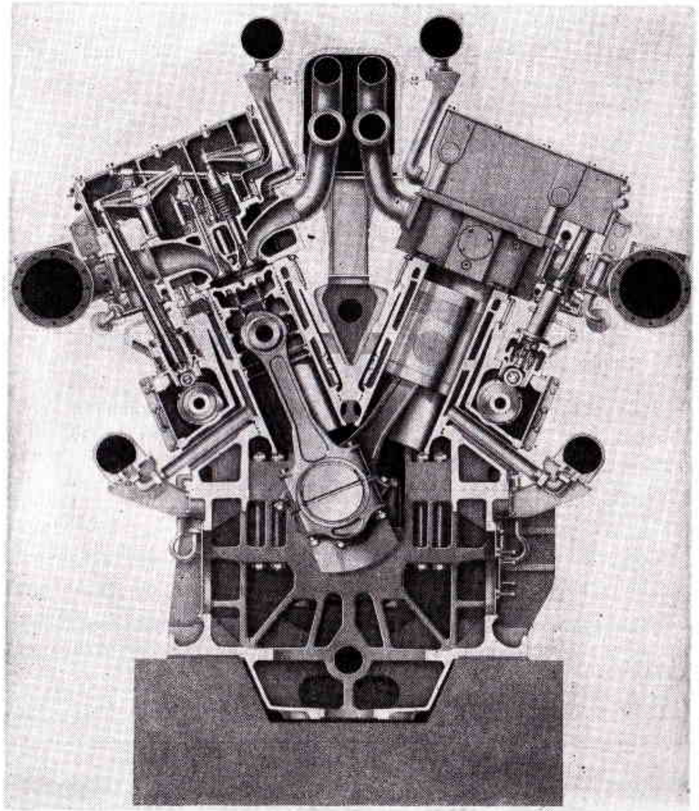


FIG. 22-6. Four-stroke diesel-engine construction. (*Courtesy of Worthington Pump and Machinery Corporation.*)

22-7. Engine Structures. Table 22-2 lists the principal components of engines and the materials of which they are made. Figures 22-5 and 22-6 show how these parts are related in two specific engines. These are cross sections through the cylinders of a typical two- and four-stroke diesel engine. Figure 22-7 shows a typical multicylinder engine used for stationary power generation.

Figure 22-8 shows an opposed-piston engine that produces a high power output for the volume of the engine. All engines need auxiliary equipment for: (1) fuel admission, (2) air admission, (3) ignition, (4) lubrication, and (5) governing.

22-8. Fuel Admission. A fuel system must supply the correct amount of fuel to each cylinder in advance of the power stroke, even with varying load. Various types of systems are available; their arrangement depends on type of fuel, compression ratio, method of ignition, and the number of strokes per cycle. The fuel is either mixed with air outside the cylinder before the intake stroke, or it is forced into the cylinder toward the end of the compression stroke. Outside mixing systems use:

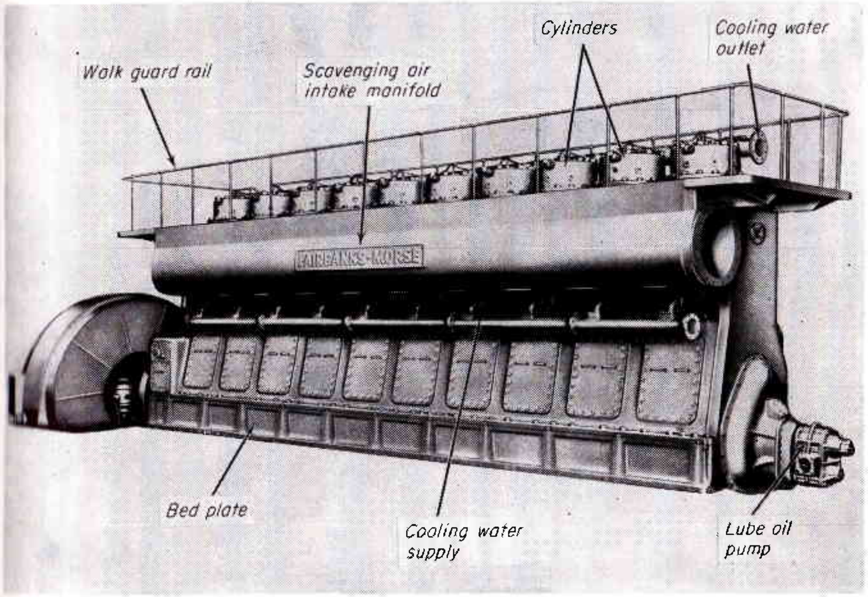


FIG. 22-7. Engine-room appearance of typical diesel-type engine. (*Courtesy of Fairbanks, Morse & Co.*)

Carburetor. To atomize liquid gasoline into the entering air stream during the intake stroke.

Gas-mixing valve. To mechanically mix the gas fuel with air entering during the intake stroke of the engine.

Inside mixing systems for oil and c-i gas engines use *injection* devices to force liquid or gas fuels into the cylinders against considerable air pressure. The injection device or system must do three things: (1) meter the amount of fuel to carry the load, (2) inject the fuel at the proper time, and (3) spray the atomized fuel to get a uniform air-fuel mixture in the cylinder.

There are two classes of injection systems: (1) mechanical-injection

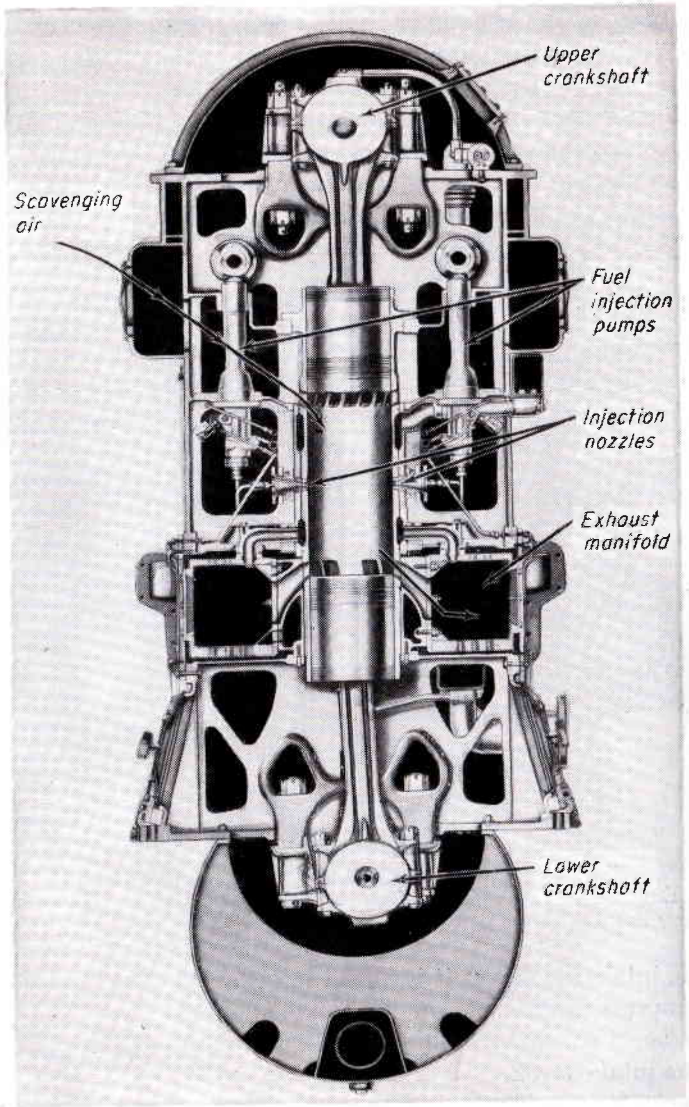


FIG. 22-8. Two-stroke-cycle opposed-piston diesel engine. Upper and lower drive shafts are connected by a vertical drive shaft. (*Courtesy of Fairbanks, Morse & Co.*)

systems using a high-pressure pump and (2) air-injection systems using high-pressure air to carry in the fuel. Mechanical-injection systems have superseded the earlier air-injection systems.

In the air-injection system a low-pressure metering pump delivers a measured amount of fuel into the cylinder fuel nozzle when the nozzle valve is closed. As the nozzle valve opens, a multistage compressor

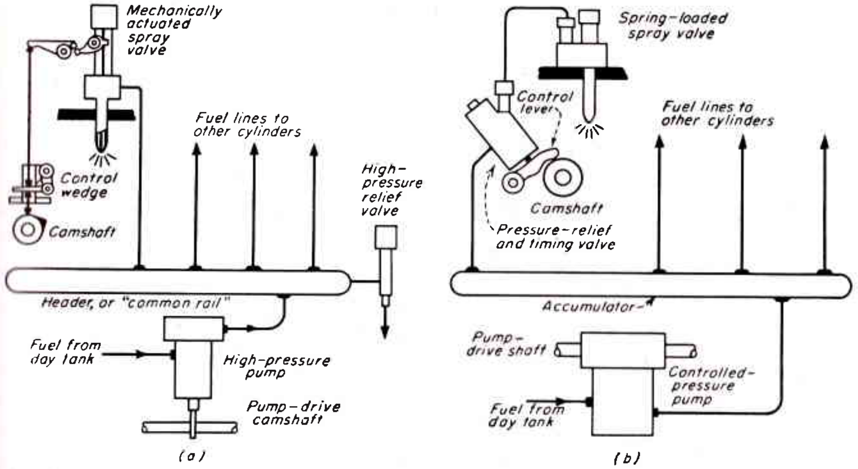


FIG. 22-9. Typical common rail injection systems for a diesel engine. A high-pressure header is supplied by a single pump with pressure held constant by a relief valve. The amount and time of injection are set by a control wedge, which adjusts the lift of a mechanically operated valve. (*Courtesy of Power.*)

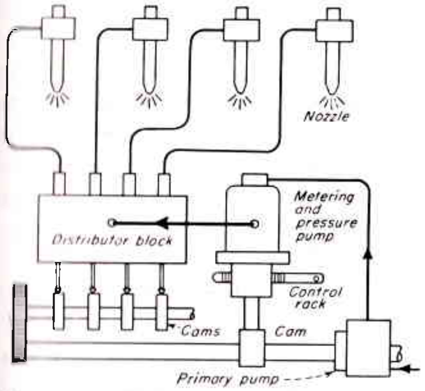


FIG. 22-10. Typical distributor injection system. A single pump meters the fuel, puts it under pressure, and times the injection. In the distributor block, cam-operated poppet valves feed fuel to the cylinders in proper firing order by opening just before injection. (*Courtesy of Power.*)

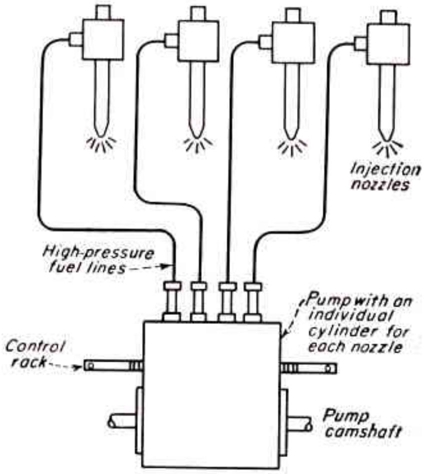


FIG. 22-11. Typical pump- and pressure-operated nozzle system. The pump system meters the fuel and controls injection timing. The nozzle has a delivery valve operated by fuel-oil pressure. (*Courtesy of Power.*)

delivers air to the nozzle to inject and spray the fuel into the engine cylinder.

Mechanical-injection systems are available in three types: (1) the common rail system, (2) the distributor-injection system, and (3) pump- and pressure-operated nozzle systems (Figs. 22-9 to 22-11). The last is the most often used.

These systems must have pumps with close clearances between plunger and barrel and must handle very small amounts of oil. The pumps are either (1) constant-stroke or (2) variable-stroke. Figure 22-12 shows a constant-stroke pump with a helical indentation on the plunger for fuel control. Other pumps use spill valves controlled by the governor to meter the fuel. Variable-stroke pumps use: (1) a governor-controlled wedge between cam and pump plunger, (2) a variable-displacement cam acting on the plunger, and (3) the varying relation of plunger and pump barrel.

Fuel nozzles, also called *spray valves,* receive the high-pressure oil from the fuel pumps and discharge it as a spray into the compressed air. Figure 22-13 shows how the oil pressure opens the nozzles during the injection period. The shapes of the orifice openings control the spray pattern, which must be selected to work with the combustion-chamber shape. Some engines use a *unit injector,* a nozzle and high-pressure fuel pump combined in a single unit.

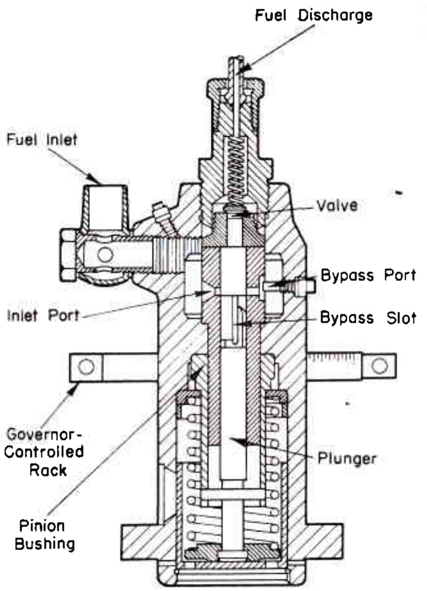


FIG. 22-12. Typical constant-stroke pump. The plunger has a helix cut into it. This helix controls the opening and closing of ports for variable fuel charges. A governor-controlled rack turns the plunger in the pump barrel. (*Courtesy of Power.*)

22-9. Special Fuel-admission Systems. A gas engine or gas diesel uses an injection nozzle that receives gas fuel pressurized in a multistage compressor. A nozzle valve controls the gas discharge into the cylinder. A small amount of fuel oil, called *pilot injection oil,* is pumped into the nozzle with the gas charge, and this helps ignite the gas-air mixture in the cylinder.

Dual-fed engines need two separate fuel-admission systems, one for gas and the other for oil. The oil may contribute anywhere from 5 to 100 per cent of the heating energy. The smaller flow acts as pilot oil for ignition, while the higher flows develop part or all of the engine output. The gas may enter the air stream directly or through a poppet valve in the cylinder, depending on whether the engine is two- or four-stroke.

22-10. Air Charging. Air enters a four-stroke engine by the aspirating action of the suction stroke through mechanically opened intake valves. Air intake may be from the atmosphere or may be delivered by an air pump. A pump delivers pressurized air. This is *supercharging;* it permits burning a larger fuel charge per cycle, yielding more work per cylinder because of the greater mass of air inducted into the cylinder of fixed dimensions.

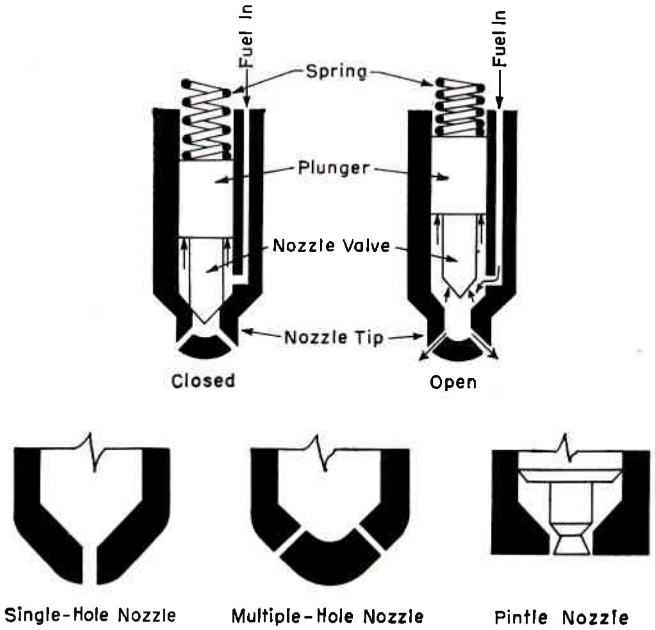


FIG. 22-13. Principles of typical injection fuel-nozzle construction are the use of a fuel-oil pressure-actuated valve and suitable orifice openings. *Single-hole* nozzles have large openings and deliver conical jets where the combustion-chamber shape creates turbulence; *multiple-hole* nozzles give a wider spray used in open-chamber designs; *pintle* nozzles may deliver a range of spray patterns from tapered to cylindrical depending on pintle design.

The exhaust stroke rids the cylinder of most of the combustion gases. In two-stroke cycles air under slight pressure flows into the cylinders through fixed port openings or mechanically controlled valves. A piston stroke does not aspirate the air. The slight positive pressure of the inlet air may be developed by a centrifugal blower, the underside of the power piston, or a separate pump. Figure 22-14 shows a rotary blower supplying air to an engine. As air enters a two-stroke cycle, it sweeps out the combustion gas in the cylinder from the previous cycle. This is called *scavenging* the gases.

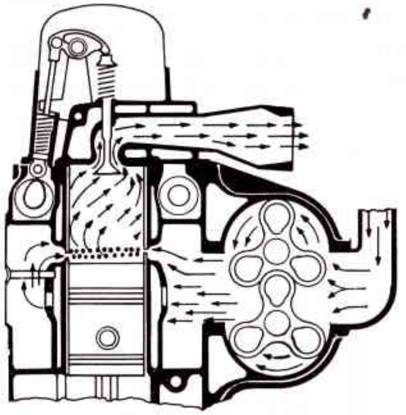


FIG. 22-14. Blower for air admission to two-stroke-cycle engine scavenges combustion gas as air is forced in.

22-11. Compression-ignition Combustion Chambers. The clearance volume of an engine confines the combustion process at the end of the compression stroke. The design of this space, called the "combustion chamber," affects the efficiency and completeness of combustion. In most engines the fuel injects directly into the cylinder or a chamber connected to the cylinder. These are classified as: (1) open chambers, (2) precombustion chambers, (3) turbulence chambers, (4) air cells, and (5) energy cells.

Figures 22-15 to 22-17 show the principle of operation of these designs. The open chambers use the shape of the piston-head inlet port and valves to create turbulence. These are good for low speeds. For higher speeds air cells store high-pressure air that blows back into the cylinder to create turbulence. The energy cell receives air, and the fuel sprayed in burns in the cell to let the combustion products blow back into the cylinder and work on the piston.

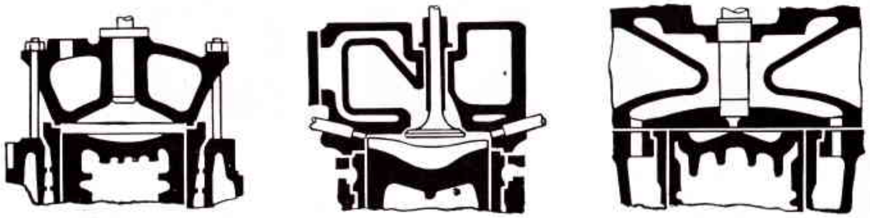
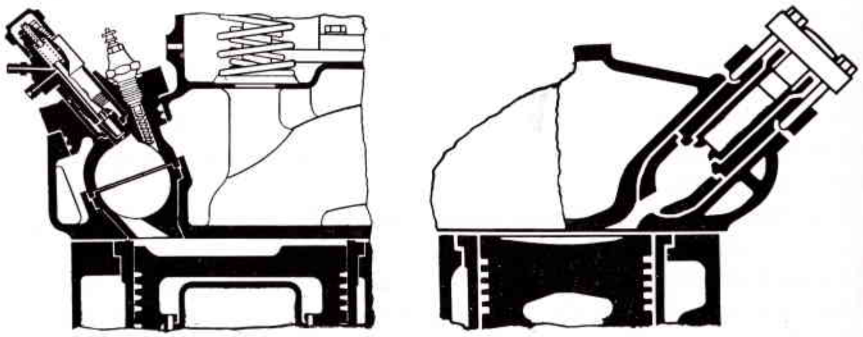


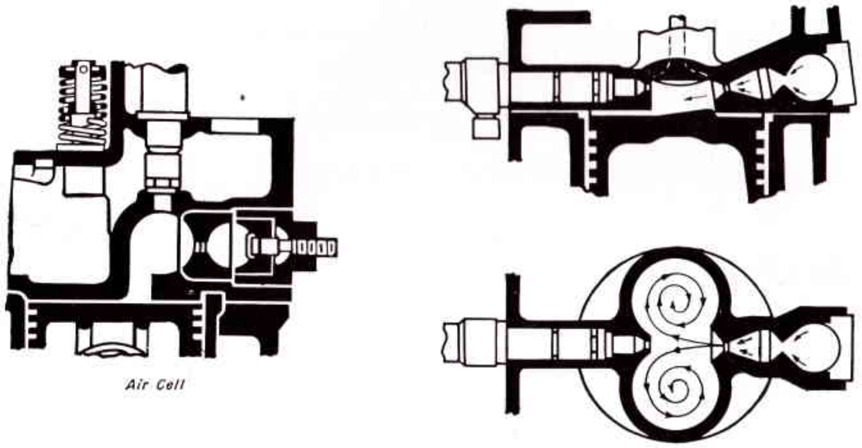
FIG. 22-15. Open-combustion-chamber designs for slow-speed engines.



Turbulence Chamber

Precombustion Chamber

FIG. 22-16. Typical turbulence chamber and precombustion chamber for high-speed diesels.



Air Cell

Energy Cell

FIG. 22-17. Typical air and energy cells for high-speed diesels.

22-12. Engine Performance. Internal-combustion engine parameters include: piston speed, air-fuel ratio, compression ratio, spark advance, and inlet-air pressure and temperature. Engines use many combinations of these, and *performance* gives the power and efficiency an engine develops as these parameters and the load vary. Performance factors may be computed as

$$V_p = 2Ln \tag{22-5}$$

where V_p = piston speed, ft per min (average)
L = piston-stroke length, ft
n = engine speed, rpm

$$V_d = LAnN \tag{22-6}$$

where V_d = engine displacement volume, cu ft or cu in.
L = piston stroke, ft or in.
A = piston area, sq ft or sq in.
n = cycles per min per cylinder
N = number of cylinders

$$bmep = \frac{bhp \times 33{,}000}{L \times A \times n} \tag{22-7}$$

where $bmep$ = brake mean effective pressure, psi
bhp = net delivered power per cylinder, hp
n = cycles per min for one cylinder
L = stroke length, ft
A = piston area, sq in.

$$imep = \frac{bmep}{\eta_m} \tag{22-8}$$

where $imep$ = indicated mean effective pressure, psi
η_m = mechanical efficiency, fraction

$$fmep = imep - bmep \tag{22-9}$$

where $fmep$ is friction mean effective pressure.

$$fhp = ihp - bhp \tag{22-10}$$

where fhp is friction horsepower.

$$HR = \frac{w_f \times HHV}{hp} \tag{22-11}$$

where HR = heat rate, Btu per hp-hr
w_f = fuel used, lb per hr
HHV = fuel heating value, Btu per lb
hp = engine output, hp (may be ihp or bhp)

$$\eta_t = \frac{2{,}545}{HR} \tag{22-12}$$

where η_t is thermal efficiency (may be based on brake horsepower or indicated horsepower).

$$sfc = \frac{w_f}{hp} = \frac{HR}{HHV} = \frac{2{,}545}{\eta_t \times HHV} \tag{22-13}$$

where sfc is specific fuel consumption in pounds per horsepower-hour (brake horsepower or indicated horsepower).

Figures 22-18 and 22-19 show the general shape of various performance factors for s-i and c-i engines. Figure 22-20 shows a performance map for an engine which gives complete performance in one set of curves. It may also be used to describe a series of similar engines. Specific engines vary in the exact shape of the contour lines they have on a performance map. All engines have a closed contour line for minimum brake specific fuel consumption, surrounded by others with progressively higher values.

22-13. Engine Developments. Internal-combustion engines compete with utility power for many industrial uses. Engine builders have progressively developed more efficient engines. Supercharging, started during World War II, was commercially applied in 1945. Four-stroke

diesels using an exhaust-gas supercharger or turbocharger gave about a 50 per cent increase in power for the same sized engine and boosted mep from about 80 to 120 psi. Fuel economy improved about 10 per cent. The four-stroke supercharged engine is standard today. In the last few years higher supercharging has boosted brake mep to over 180 psi with a further reduction in specific fuel consumption.

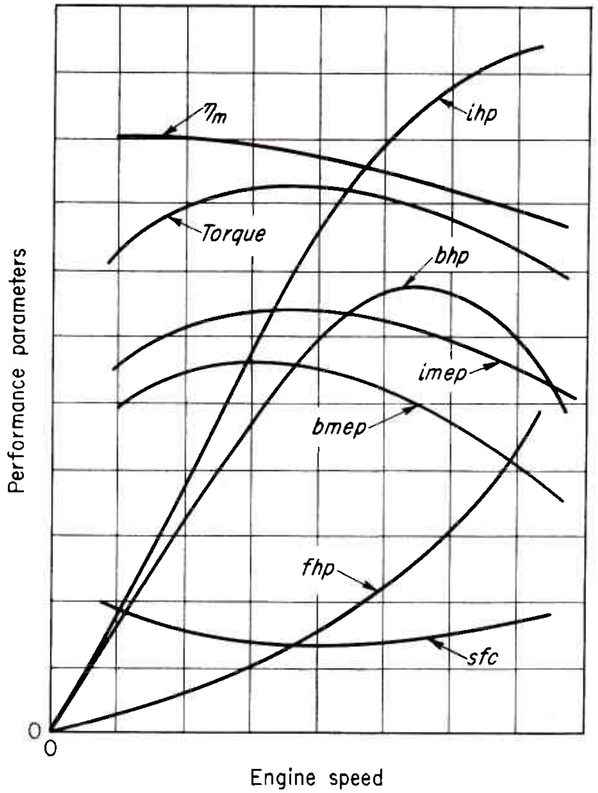


FIG. 22-18. Performance trends for an s-i engine. These curves apply to *full-open-throttle* operation.

About 1937 the two-stroke gas diesel was introduced, with gas injection at high pressure and ignition controlled by the injection of pilot oil. About 1946 this design appeared as a four-stroke gas diesel operating on the high-compression diesel cycle followed by supercharging of the gas unit. In 1949 the four-stroke dual-fuel engine with either atmospheric or supercharged intake appeared. The engine may operate on either gas or oil. About the same time the s-i high-compression supercharged gas engine was developed. Table 22-3 shows the approximate performance of the newer engines.

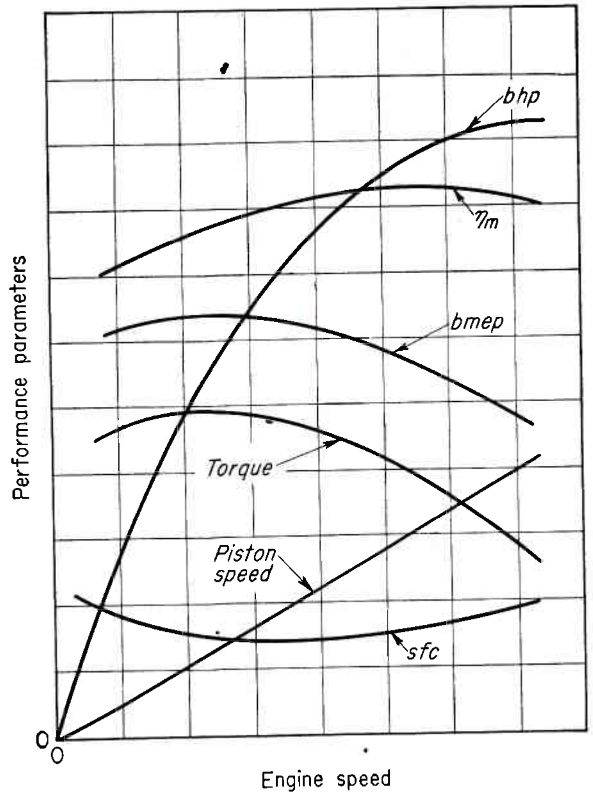


FIG. 22-19. Performance trends for a c-i engine.

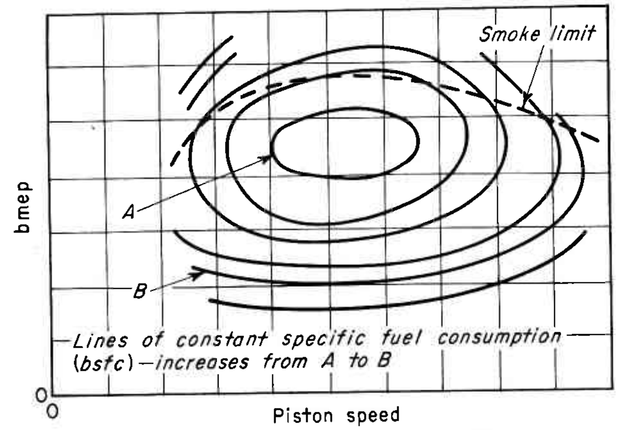


FIG. 22-20. Typical performance map for a c-i engine. This type of map appears much the same when plotted for s-i engines.

TABLE 22-3. APPROXIMATE PERFORMANCE DATA—COMPRESSION-IGNITION AND SPARK-IGNITION ENGINES

Type engine	Fuel	Compression	Ignition	Approx. full-load heat rate, Btu per bhp-hr
Gasoline engine	Gasoline	Low	Spark	12,000
Gas engine	Natural or other gas	Low	Spark	10,000
Oil engine (semidiesel)	Oil	Low	Hot spot	9,000
Oil engine (diesel)	Oil	Medium	Spark	7,500
Oil engine (diesel)	Oil	High	Compression	7,200
Gas-diesel engine	Mainly gas	High	Pilot oil	7,200
Dual-fuel engine (nonsupercharged)	Gas or oil	High	Pilot oil and/or compression	7,200
Gas engine (supercharged)	Gas	High	Spark	7,150
Dual-fuel engine (supercharged)	Gas or oil	High	Pilot oil and/or compression	7,100
Oil engine (diesel, supercharged)	Oil	High	Compression	6,900

About 1950 attention was given to temperature control by cooling the combustion air to increase power output of the engine. Air precoolers and aftercoolers were designed for supercharged engines. Cooling intake air has raised brake mep to 200 psi. In addition lower fuel consumption, better combustion, and improved engine life were achieved.

During 1954 supercharging was applied to two-stroke engines; Table 22-4 summarizes the development trends in brake mep.

TABLE 22-4. TREND IN INTERNAL-COMBUSTION-ENGINE BRAKE MEAN EFFECTIVE PRESSURE

Year	Four-stroke engine bmep, psi	Two-stroke engine bmep, psi
1940	75–80[a]	65–68[a]
1945	120[b]	70[a]
1950	140[b]	78[a]
1953	160[b]	80[a]
1954	180[b]	87–100[b]
1956	200[b]	120[b]

[a] Nonsupercharged.
[b] Supercharged.

The Supairthermal engine appeared several years ago; it uses high-pressure supercharging at about 15 to 20 psig. A special mechanism automatically controls the closing of the intake-air valve. This permits expansion and cooling of the air during the suction stroke. More fuel may be burned with higher output per cylinder.

Manufacturers are turning also to the V type of cylinder arrangement, which allows almost double the power output in a given space. Engines today are also designed for better cooling and lubricating systems with better materials for engine parts. Table 22-5 shows the trend in engine speeds.

TABLE 22-5. TREND IN INTERNAL-COMBUSTION ENGINE SPEEDS

Bore and stroke, in.	1946 (avg) speed, rpm	1956 (avg)	
		Speed, rpm	Piston speed, ft per min
Four-stroke:			
4 × 5	2,000	2,500	2,080
7 × 10	900	1,200	2,000
13 × 16½	450	600	1,650
16 × 22	300	360	1,320
Two-stroke:			
5 × 5.6	.....	1,800	1,680
8¾ × 10½	720	720–850	1,260–1,488
21½ × 31	225	240	1,240
20 × 40	164	180	1,200

PROBLEMS

22-1. A 3,000-kw diesel generating unit gave the following performance values:

Fuel rate, 1.5 bbl for 900 kwhr of 25°API fuel
Mechanical efficiency, 82.0 per cent
Generator efficiency, 92.0 per cent

Calculate:
a. Engine fuel rate
b. Engine-generator fuel rate
c. Indicated thermal efficiency
d. Over-all thermal efficiency
e. Brake thermal efficiency

22-2. The data of Table P-1 were obtained from the test of a single-acting four-stroke five-cylinder solid-injection diesel engine. The engine is rated at 700 bhp and 250 rpm, using 30°API fuel.

TABLE P-1

Bhp	Ihp	Fuel flow, lb per hr	Exhaust gas temperature, deg F
200	335	128	218
300	468	159	257
400	588	190	298
500	708	216	337
600	829	261	388
700	947	302	455
800	1,075	357	520

Plot the following performance curves:
a. Fuel flow
b. Fuel rate
c. Exhaust-gas temperature
d. Mechanical efficiency
e. Brake thermal efficiency
f. Indicated thermal efficiency

22-3. Data for a single-acting four-stroke diesel engine under test are as follows:

Fuel rate, 24 lb per hr of 24°API fuel
Prony brake, 42-in. arm, 280-lb gross, 25-lb tare
Speed, 420 rpm
Engine dimensions, 9 by 14 in.

Calculate the brake thermal efficiency.

22-4. A Froude hydraulic dynamometer with a constant of 2,000 is used for testing a 5- by 7-in. four-stroke single-acting six-cylinder diesel engine at 1,000 rpm. Data are as in Table P-2.

TABLE P-2

Load, per cent	Dynamometer reading, lb	Fuel flow, lb per hr
25	40	9.2
50	80	17.6
75	120	25.8
100	160	33.6
110	176	37.9

Plot the following performance curves:
a. Specific fuel consumption
b. Brake mep
c. Engine displacement in cubic feet per minute per brake horsepower

22-5. A 12- by 18-in. four-cylinder four-stroke single-acting diesel engine is rated at 200 bhp at 260 rpm. Fuel consumption at rated load is 0.42 lb per bhp-hr with a heating value of 18,920 Btu per lb.

Calculate:

a. Brake mep

b. Engine displacement, cubic feet per minute per brake horsepower

c. Brake thermal efficiency

22-6. A two-stroke 15⅛- by 21-in. diesel engine with four cylinders is guaranteed to deliver 700 hp at 300 rpm with a specific fuel consumption of 0.42 lb per bhp-hr based on 19,050 Btu fuel. Engine is single-acting.

Calculate:

a. Brake thermal efficiency

b. Brake mep

c. Suction displacement, cubic feet per minute per brake horsepower

d. Btu supply to cylinder per cubic feet of displacement

22-7. The fuel consumption of a 2,500-hp diesel engine is shown in Table P-3.

TABLE P-3

Bhp	Fuel flow, lb per hr	Bhp	Fuel flow, lb per hr
500	235	2,000	800
1,000	410	2,500	1,012
1,500	602	3,000	1,290

The load data for week days are given in Table P-4.

TABLE P-4

Time	*Average load, hp*
7 A.M.– 5 P.M.	2,460
5 P.M.–10 P.M.	720
10 P.M.– 7 A.M.	530

The Sunday load is 200 hp.

Calculate the yearly fuel cost based on 24°API fuel oil selling at $4.95 per barrel.

22-8. A four-cylinder 16½- by 24½-in. four-stroke diesel engine has a fuel consumption of 0.38 lb per ihp-hr based on 18,960 Btu oil. Engine speed is 257 rpm with an indicated mep of 123 psi. Jacket water carries away an estimated 22 per cent of the heat supplied. Calculate the gallons per minute based on a temperature of 80 F. Engine is single-acting.

22-9. Calculate the cooling water needed in gallons per minute for a diesel with the following performance data:

Brake horsepower, 4,500
Jacket loss, 23 per cent
Water temperature rise, 60 F
Mechanical efficiency, 83.0 per cent
Indicated thermal efficiency, 34.0 per cent

22-10. A 1,500 six-cylinder 23- by 28-in. 225-rpm four-stroke diesel has a fuel consumption of 0.38 lb per hp-hr based on 19,000 Btu oil. A waste-heat-recovery boiler recovers 35 per cent of the exhaust loss. Jacket losses are estimated at 30 per cent. Estimate the quantity of 5-psig steam that can be produced in pounds per hour if jacket water at 160 F is used for boiler feed.

22-11. A two-stroke six-cylinder single-acting 14- by 14-in. gas engine is rated at 600 hp at 300 rpm. It uses natural gas with a consumption of 10,000 Btu per hp-hr.

Calculate:

a. Thermal efficiency

b. Fuel cost per hour based on 900 Btu gas at $0.60 per 1,000 cu ft

c. Purchased power cost in cents per kilowatthour equivalent to engine operating fuel cost

22-12. Table P-5 gives an abstract of test data for a gas-engine generator in a central station.

TABLE P-5

	Load, per cent					
	122.5	110	100	75	50	0
Kilowatts	1,225	1,104	1,003	733	578	0
Generator efficiency	95.3	95.2	95.1	94.7	94.2	
Cubic feet gas per hour	12,750	11,770	11,170	8,670	7,680	4,430

High heating values, 1,149 Btu per cu ft
Engine speed, 225 rpm

Plot the following performance curves against engine horsepower:

a. 10^6 Btu per hr

b. Btu per brake horsepower-hour

c. Btu per kilowatthour

d. Engine thermal efficiency

e. Over-all thermal efficiency

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CHAPTER 23

INTERNAL-COMBUSTION ENGINE PLANTS

23-1. Applications. Internal-combustion (i-c) engines of many types are used for stationary power generation in a wide variety of services: central stations, oil fields, pipelines, sewage disposal, and commercial, institutional, and military bases. Both compression-ignition and spark-ignition engines serve in steam stations to supply auxiliary power and in some industrial plants and institutions as emergency stand-by sources of energy in the event of main power-supply failure. In some smaller systems i-c engines work with steam units to supply the peak-load demands on the plant. Seasonal electric loads sometimes may be supplied more economically by an i-c engine plant.

Internal-combustion engine plants need equipment to serve the engine, such as:

Air-intake systems. Intake filters, ducts, and silencers

Exhaust-gas systems. Ducts, mufflers, water heaters, steam boilers

Fuel systems. Storage tanks, pumps, strainers, oil filters, meters, oil heaters, piping

Engine cooling systems. Pumps, heat exchangers, cooling towers, spray ponds, water treatment, piping

Lube-oil systems. Pumps, tanks, relief valves, filters, coolers, purifiers, piping

Engine starting systems. Battery, electric motor, mechanical air compressor and engine, wiring, control panel

Electrical systems. Generators, switchgear, exciters, busses, transformers, auxiliary power and light

Figure 23-1 shows some of this auxiliary equipment in schematic form.

23-2. Air-intake Systems. Air-intake systems lead air into the engine cylinders through ducts or pipes and filters. The filters remove air-borne solids that may act as an abrasive in the engine cylinders. Filter types include: (1) dry, (2) oil-bath, (3) viscous-impingement, and (4) electrostatic precipitator filters.

The intake system must cause a minimum pressure loss to avoid reducing engine capacity and raising specific fuel consumption. Filters must be cleaned periodically to prevent pressure loss from clogging. Silencers must be used on some systems to reduce high-velocity air noises.

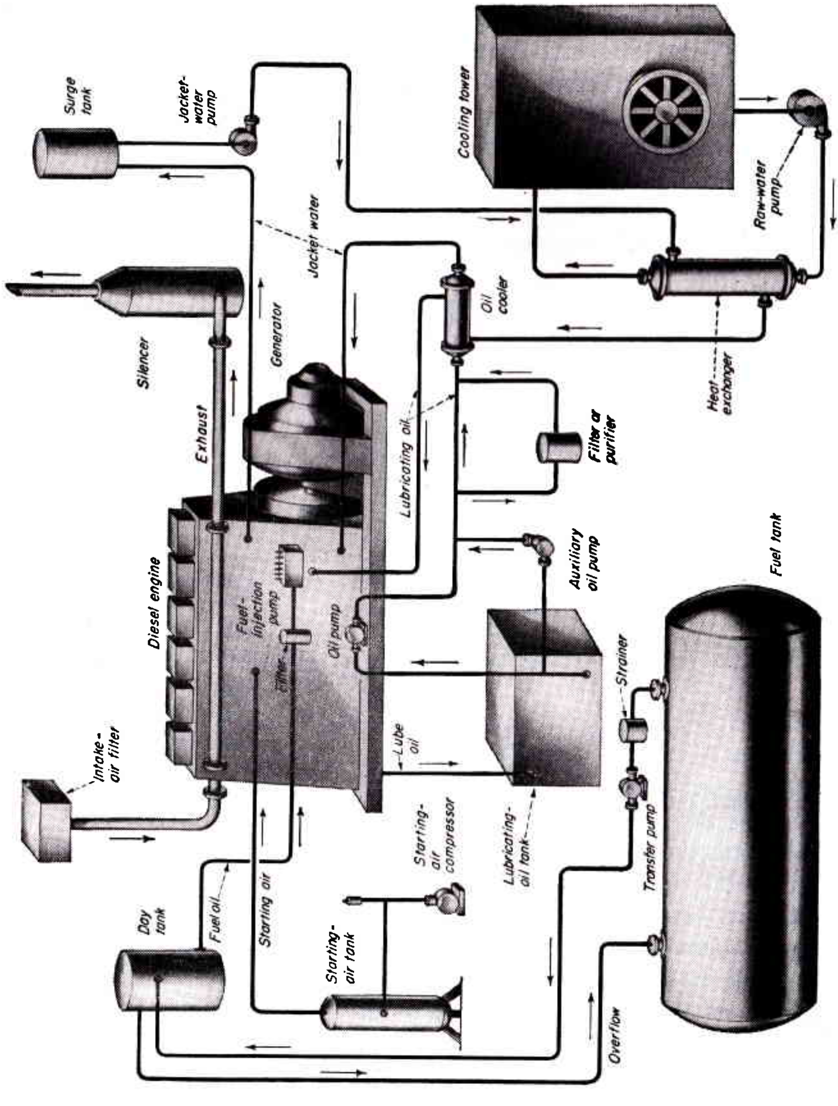

FIG. 23-1. The internal-combustion engine plant consists basically of the engine and related systems of equipment for air, exhaust, fuel, cooling lube oil, and water auxiliaries.

23-3. Exhaust-gas Systems. These systems lead the engine exhaust gas outside the building and discharge it to atmosphere. They also must be designed for low pressure loss to avoid cutting engine capacity and reducing efficiency. All exhaust systems need mufflers to attenuate gas-flow noises, which are highly objectionable. The exhaust stack usually stands on the muffler top.

Where some of the exhaust enthalpy energy may be useful, it may be recovered in gas-to-air heat exchangers or gas-to-water exchangers or in waste-heat boilers for steam generation. These devices also act as mufflers.

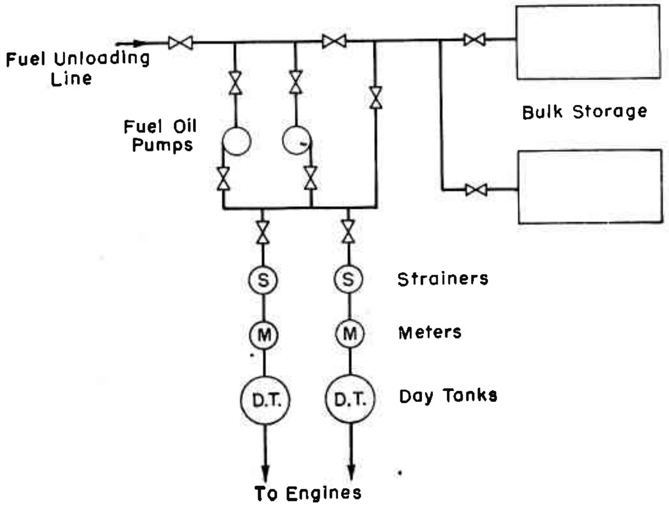

Fig. 23-2. Fuel systems have tankage and transfer systems as indicated.

23-4. Fuel Systems. *Bulk storage tanks* and *engine day tanks* hold the engine fuel oil. The former receive the oil delivered to the plant and stand outdoors for safety. Pumps draw oil from the storage tank to supply the smaller day tanks in the plant at daily or shorter intervals. Large storage capacity allows purchasing fuel when prices are low. Figure 23-2 shows a typical tank system. Oil may be delivered by truck, 600 to 5,000 gal, railroad tank cars, 6,000 to 10,000 gal, or barge and tanker, with very large capacities.

Storage tanks aboveground must be surrounded by a dike to form a moat large enough to hold the tank contents if the tank should leak. Tanks must have: manholes for internal access and repair, fill lines to receive oil, vent lines to discharge vapors, sounding connections to measure content, overflow return lines for controlling oil flow, and a suction line to withdraw oil. Coils heated by hot water or steam reduce oil viscosity to lower pumping power needs.

Delivered oil sometimes holds water, dirt, metallic chips, and other foreign matter that must be removed by filtering or centrifuging. Much of this will settle out in the storage tank, especially with the lighter fuel oils and at higher temperatures. A light degree of contamination may be removed by filters, but heavier fouling requires centrifuging.

Dip-stick measurements give quick spot checks of tank contents, but meters give continuous indications. Figure 23-2 shows flowmeters between storage and day tanks, but not between the latter and the engines. The return line from the injection systems would make such meters useless since they would measure the gross oil pumped to the injection system, but not the net amount burned by the engine. Volumetric meter readings must be corrected for oil temperature to determine the weight of oil burned.

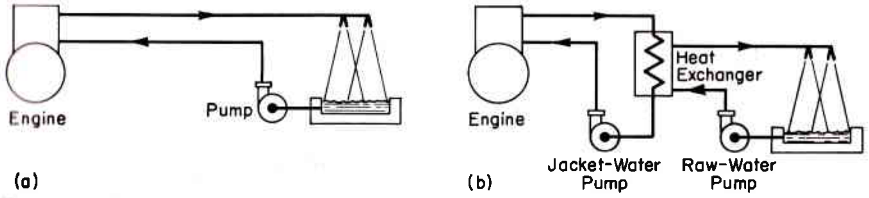

Fig. 23-3. Engine cooling systems are either (*a*) open- or single-circuit type; (*b*) closed or double circuit.

23-5. Engine Cooling Systems. Most of the thermal energy supplied to engines leaves the engine in thermal form: (1) as energy in the high-temperature exhaust gases and (2) as heat transferred to the jacket cooling water. Removing the latter heat prevents damaging the cylinder liners, heads, and walls and the piston and its rings. Small engines may be air-cooled, but larger stationary engines use water circulating in cylinder jackets.

Cooling water must be controlled in temperature: when too low, the lube oil will not spread properly and the cylinder and piston wear; when too high, the lube oil burns. Small-diameter cylinders have leaving temperatures up to about 180 F; large-diameter cylinders use lower leaving temperatures.

Constant cooling-water flow rate makes jacket-temperature rise vary with the load when inlet temperature stays constant.

Figure 23-3 shows two types of cooling-water systems. The single-circuit system may be subject to corrosion in the cylinder jackets because of the dissolved gases in the cooling water. The double-circuit system largely eliminates internal jacket corrosion but may have corrosion in the raw-water circuit of the heat exchanger.

When large bodies of water are nearby, they may be used for jacket cooling in i-c engines. Most plants, however, use a spray pond, cooling tower, or evaporative cooler. These can cool the water close to the wet-bulb temperature of the air, within about 10 to 18 F. Figure 23-4 shows the variety of towers used by i-c engine plants.

Systems using cooling towers usually treat the make-up water, depending on the water contaminants. They use zeolite softeners and sometimes lime or the lime-soda-ash treatment.

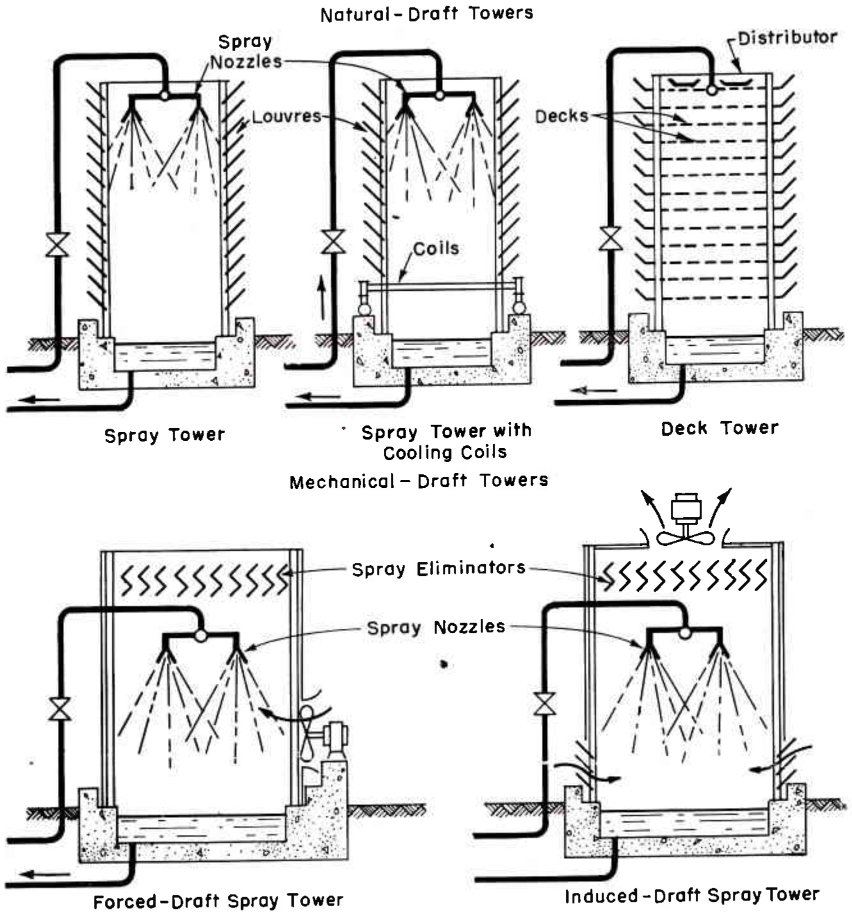

FIG. 23-4. Cooling towers operate basically as *natural*-draft or *mechanical*-draft towers. (*Courtesy of Power.*)

23-6. Lubricating-oil Systems. The lubricating oil, or *lube oil*, performs several duties: (1) it lubricates moving parts, (2) removes heat from cylinder and bearings, (3) helps piston rings to seal gases in the cylinder, and (4) carries away solid matter from rubbing moving parts. Lube oils must be chosen with care and purified at intervals.

Modern lube oils have additives to act as: oxidation inhibitors, foam-reducing agents, pour-point depressants, and other agents. Dopes and additives may be used in oils to refresh them.

Plant operators choose oils of the grade recommended by the engine builder and from reputable oil refiners, mainly from the actual performance of the oil in service rather than from information based on laboratory tests. Using several brands with possible different additives may cause trouble. Oil in service becomes contaminated with dust, atmospheric moisture, dirt, carbon, and metallic chips. Heat causes chemical changes in the oil. Oil must be used as long as possible to keep costs down; purifying extends its life. Temperatures above 225 F shorten lube-oil life.

Lube oil is purified by four methods: (1) settling, (2) centrifuging, (3) filtering, and (4) chemical reclaiming. Centrifuging, widely used, gives excellent purification when properly done. Filters are made as: (1) strainers, (2) absorbent filters, and (3) adsorbent filters. They effectively remove small amounts of impurities but are costly for large amounts.

Chemical reclaiming uses a combination of heat and activated clay followed by filtering. Effective settling methods need large heated tanks; the clean oil is drawn from the top.

23-7. Engine Starting Systems. Starting systems include: (1) air starting for medium and large-capacity stationary and mobile units, (2) auxiliary-engine starting for medium-capacity mobile units, and (3) electric-motor starting for small high-speed gasoline and diesel engines.

Air starting uses valve arrangements to admit pressurized air, about 250 psi, to a few of the cylinders, making them act as reciprocating air motors or engines to turn over the engine shaft. Admitting fuel oil to the remaining engine cylinders makes the engine start under its own power. A gas-engine-driven compressor usually supplies air to the compressed-air starting tanks.

23-8. Waste-heat Recovery. Thermal energy in exhaust gas and jacket-cooling water may supply heat for a variety of purposes, such as for heating fuel oil, for space heating, for hot water, or for generating process steam. Recovery arrangements include: (1) circulating air around the exhaust system, (2) using hot water from the engine jackets, and (3) using waste-heat boilers in mufflers to provide steam or hot water.

23-9. Sewage-treatment Plant. These plants treat sewage to speed up the natural processes of fermentation and purification. Engines provide shaft power and may be of a wide variety of types.

Larger treatment works are beginning to use trifuel engines. These can function as dual-fuel units. Removing plugs in the head and replacing them with spark plugs converts them to high-compression s-i units.

The Bay Park sewage-treatment works of Nassau Co., N.Y., illustrates how power and process heat are used. Figure 23-5 shows the phases

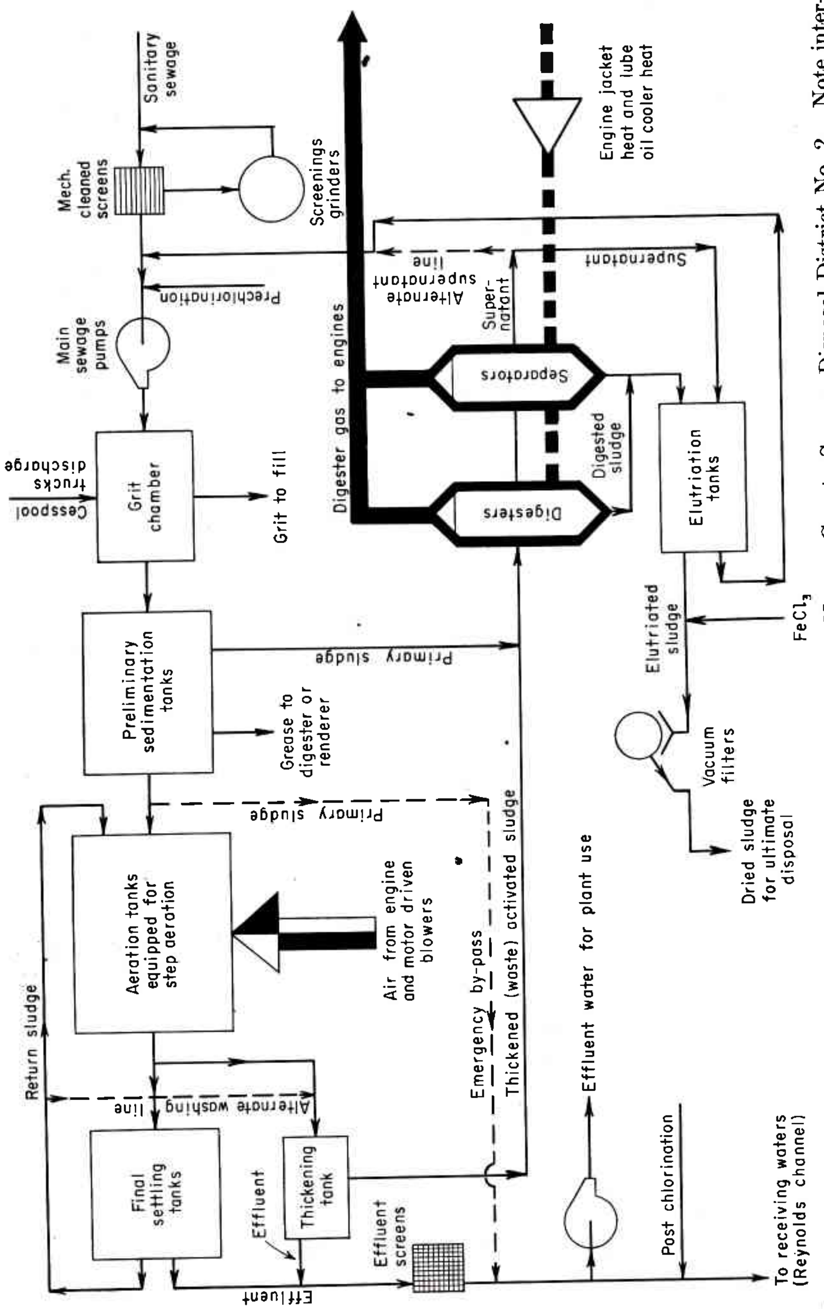

FIG. 23-5. Flow sheet of the Bay Park sewage-treatment works, Nassau County Sewage Disposal District No. 2. Note interrelationship of treatment equipment and services from an associated combustion engine plant. (*Courtesy of Department of Public Works, Nassau Co., N.Y.*)

of the sewage-treatment cycle and the major components. Liquid wastes enter the plant by gravity flow through an 84-in.-diameter trunk sewer. In the first treatment phase the raw sewage passes through a vertical, mechanically raked bar screen to remove materials damaging to pumps and equipment. The screened sewage flows to the suction of three 37-mgd pumps, one 30-mgd pump, and one 20-mgd pump. Any combination of these pumps forces the sewage through a venturi meter to the grit chamber.

This removes the inorganic materials such as sand and cinders. This grit can be used for fill. The sewage flows from the grit chamber by gravity to the preliminary sedimentation tanks, where the heavy organic solids settle out after a 1-hr retention, forming a sludge blanket. Mechanical scraping equipment continuously removes the sludge. On the return trip the partly submerged scraping equipment skims the floating kitchen wastes and other matter.

Sludge and skimmed material go to the digester via plunger pumps. Sewage flows to the aeration tanks and contacts a culture of aerobic organisms. Diffusers introduce fine air bubbles into this mixture. The air (1) mixes the culture with settled sewage, (2) prevents sedimentation of the solids undergoing aeration, and (3) supplies some of the oxygen needed by the organisms. Air consumption varies from 0.5 to 1.2 cu ft per gal of sewage. Power to supply this air is 60 to 70 per cent of the total power needs of the plant.

After about 5 hr aeration the mixture flows into the final settling tanks to be retained some 2.2 hr. Solids containing helpful aerobic organisms settle out, and after mechanical removal they are pumped back to aeration tanks to mix with the incoming settled sewage. To maintain a balance of solids, part of the aerated mixture goes to the thickening tanks, where solids compact and are then pumped to the digester.

Clear liquid overflowing from the final settling tanks is screened through ⅜-in.-square openings. Chlorine disinfects the effluent that flows through a 2.5-mile pipe to the receiving waters. Several million gallons of this chlorinated sewage is used daily at the plant for cooling water in heat exchangers, landscape watering, foam-control sprays, etc.

The various solids fed to the digesters undergo intensive anaerobic bacterial action for 20 to 30 days at 90 F. Some solids break down and convert to gases; others stabilize and become inoffensive. The gas, principally methane, is used as fuel in the dual- and trifuel engines in the plant. Engine jacket-water heat maintains the digester temperature. Solids from the digesters go to the separator to separate liquid from the solids which are compacted.

The elutriation tanks water-wash the sludge to remove certain ammoniacal compounds which interfere with filtration. The sludge is con-

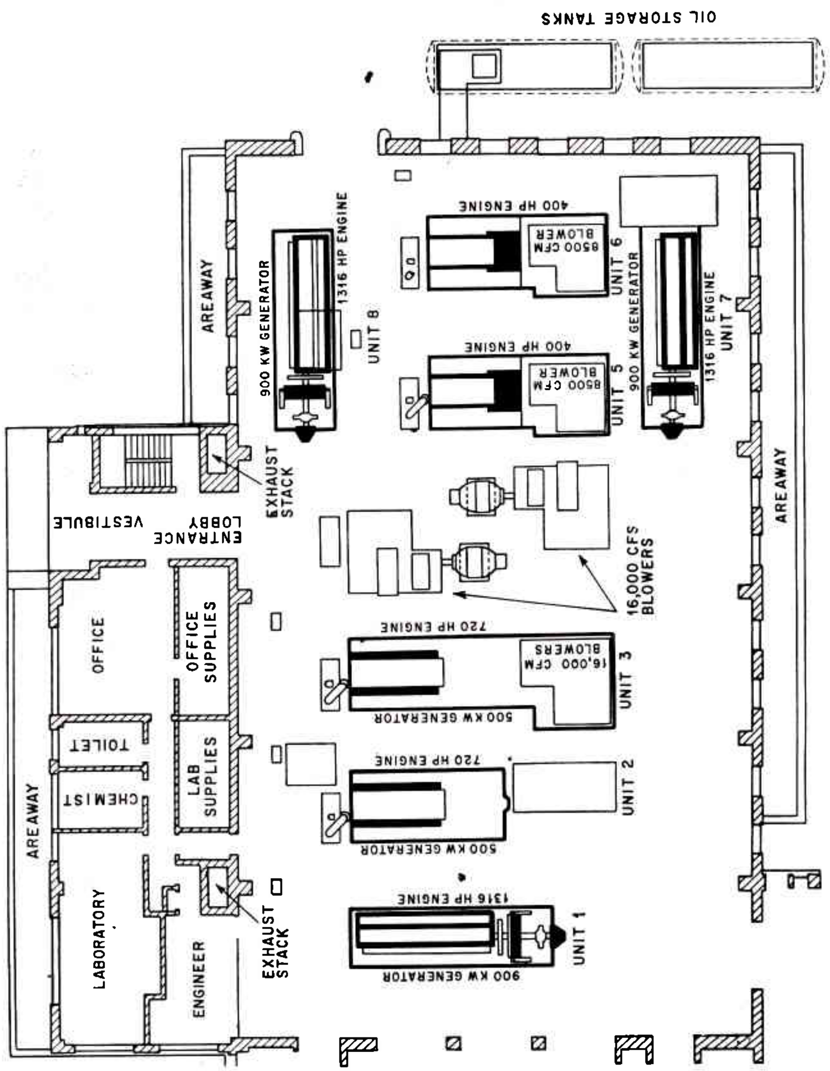

FIG. 23-6. Engine-room floor plan of Bay Park sewage-treatment works. (*Courtesy of Department of Public Works, Nassau Co., N.Y.*)

ditioned with ferric chloride and filtered on the surface of a continuous-type vacuum filter. The resulting sludge cake (75 per cent moisture) can be used as soil conditioner or fertilizer base or for land reclamation.

Sludge gas, about 65 per cent methane, has a high heating value (HHV) of 550 to 600 Btu per cu ft. In this plant the sulfur content of the gas is less than 1 grain per 100 cu ft. High sulfur, 20 to 200 grains, needs scrubbing.

The i-c engines supply all the plant's power needs; no utility stand-by service is used. Figure 23-6 shows the plant layout, using four-cycle turbocharged engines:

Units No. 1, 7, *and* 8. Worthington Pump and Machinery Corporation—6-cylinder, 4-cycle, supercharged, heavy-duty, trifuel, 1,316 bhp at 360 rpm, 16 by 20 in., piston speed 1,200 ft per min, 120 psi brake mep, equipped with intercoolers for air from superchargers; engines drive 900-kw 40°C-rise generators. Intercoolers plus 40°C-rise generators increase unit capacity 15 per cent.

Unit No. 2. Worthington Pump and Machinery Corporation—7-cylinder, 4-cycle, 720 bhp at 514 rpm, dual-fuel, direct-connected to 625-kva 500-kw 0.8-power-factor a-c generator with a 7.5-kw 125-volt direct-connected exciter.

Unit No. 3. Worthington Pump and Machinery Corporation—6-cylinder, 4-cycle, 720 bhp at 360 rpm, direct-connected to 625-kva 500-kw 0.8-power-factor a-c generator with a 15-kw 125-volt, 1,150-rpm multi-V belt-driven exciter, connected by a special flexible coupling to a 32- by 35½-in. blower with a capacity of 16,000 cfm.

Unit No. 4 (located in another building). Worthington Pump and Machinery Corporation—6-cylinder, 4-cycle, 615 hp at 514 rpm, direct-connected to 500-kva 400-kw 0.8-power-factor a-c generator with a 7.5-kw 125-volt direct-connected exciter.

Units No. 5 *and* 6. Worthington Pump and Machinery Corporation—5-cylinder, 4-cycle, 400 bhp at 400 rpm, direct-connected through flexible coupling to a 26- by 27½-in. blower with a capacity of 8,500 cfm at 360 rpm.

All engines have waste-heat boilers in their exhaust systems. The boilers on Units No. 1, 7, and 8 develop 15 psi steam for building and tunnel heating. The lubrication system uses straight mineral napthenic-base oil. Older engines use fuller's earth bypass filters; the newer ones use full flow. A portable centrifuge quickly cleans lube oil in each crankcase. Weekly oil samples check conditioning needs.

Table 23-1 gives the principal statistics for the plant.

23-10. Internal-combustion Engine in Steam Plant. In these arrangements the i-c engine is not a main unit but usually an emergency stand-by or peak-load generator. *Diesel Power* in the February, 1957, issue describes a dual-fuel engine in the Woodstock, Ill., Municipal Steam Plant.

The plant has no tie with another utility system and uses the i-c engine for emergency stand-by. The engine is a Nordberg Supairthermal dual-fuel engine rated at 3,660 bhp at 514 rpm. It works with three steam-turbine generators, totaling 14 mw capacity.

The engine originally intended for stand-by service can be put on the line in a few minutes and can supply the total residential load. Stand-by turbines take 1 to 2 hr to place in service. The new unit can also be started quickly to meet unexpected peak loads. When natural gas is burned in the summer, its thermal efficiency warrants running it on base load, since dump-gas rates are low in cost.

TABLE 23-1. APPROXIMATE PLANT DATA OF BAY PARK SEWAGE-TREATMENT WORKS

Plant item	*Value*
Sewage treated, mgd:	
Present	27
Proposed	60
Population served:	
Present	275,000
Proposed	600,000
Original plant cost (total)	$7,450,000
Power-plant equipment	$ 600,000
Total power produced, adjusted for engine-driven blower:	
Present	8.4 × 10⁶ kwhr per year
Proposed	17.2 × 10⁶ kwhr per year
Maximum demand, kw (proposed)	3,160
Sewage-sludge gas, cu ft per year (present)	90×10^6
Fuel oil, gal per year (present)	220,000

23-11. Internal-combustion-engine Municipal Plant. Internal-combustion engines are widely used in small municipal power plants. The plant in Cushing, Okla., is typical; it serves a load made up of refineries, oil and gas companies, oil-well servicing companies, oil drillers, meat-packing plants, the retail business district, and a domestic load of 12,000 population. The Cushing Plant installed its first three 750-hp McIntosh & Seymour diesels in 1935, an 1,170-hp McIntosh & Seymour engine in 1939, and a 1,420-hp Fulton engine in 1948. In recent years association with a utility system helped to meet a growing load.

In 1956 the plant was expanded by: (1) adding a 16-cylinder turbocharged Cooper Bessemer dual-fuel unit rated at 3,560 hp at 327 rpm and driving a 2,500-kw generator; (2) building a plant addition and installing two Fairbanks Morse dual-fuel engines, each of 10 cylinders, 18-in. bore, and 27-in. stroke and rated at 3,500 hp at 277 rpm driving a 2,500-kw

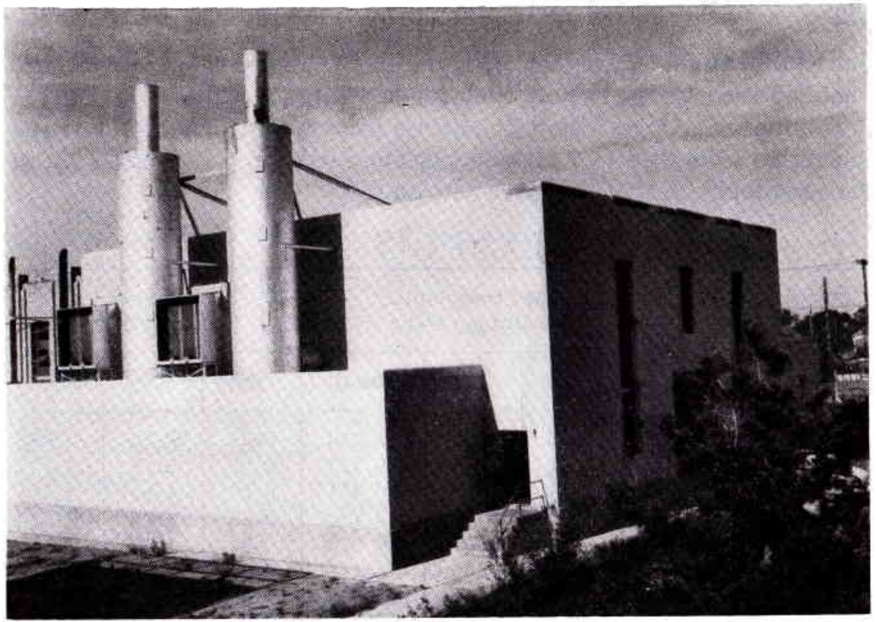

FIG. 23-7. New section of the Cushing Municipal Power Plant houses two 3,500-hp Fairbanks Morse dual-fuel units, major part of an expansion program that tripled capacity of the plant. The view shows the Maxim exhaust silencers and American air filters. *(Courtesy of Fairbanks, Morse & Co.)*

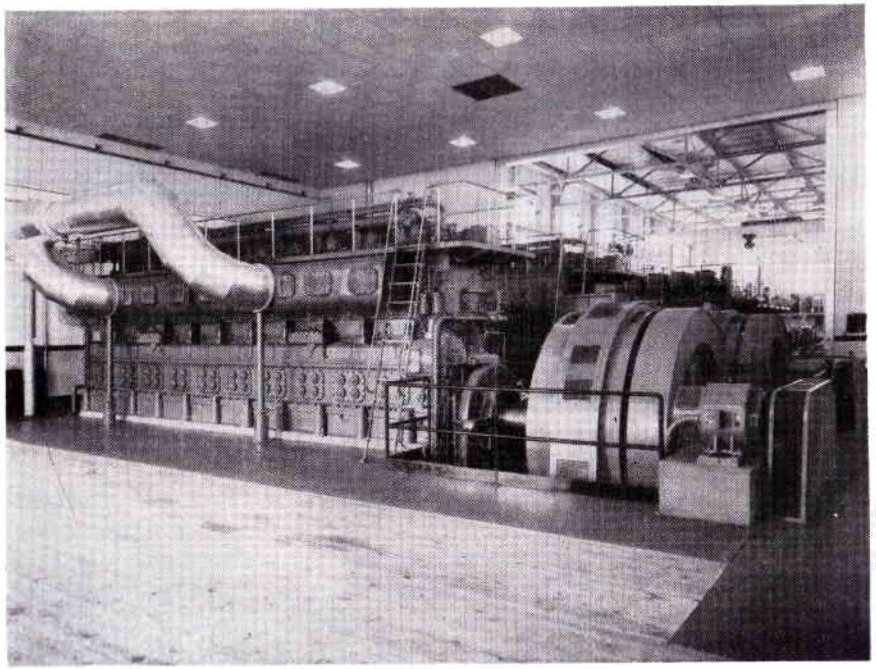

FIG. 23-8. Looking from the new section into the old section of the Cushing Plant, one sees the eight dual-fuel diesels with total capacity of 15,400 kw. The two 3,500-hp units in the foreground are in the new section. *(Courtesy of Fairbanks, Morse & Co.)*

Fairbanks Morse alternator. Figure 23-7 shows the extension and Fig. 23-8 the engines, looking from the new into the old section.

The fuel economy of the new engines has reduced the costs of power generation from 3.53 to 2.41 mills per kilowatthour. Table 23-2 shows the acceptance tests of the three new units.

TABLE 23-2. ACCEPTANCE-TEST RESULTS, CUSHING, OKLA.

Unit No.	% load	Fuel consumption, Btu per kwhr			Fuel cost, mills per kwhr		
		Pilot	Gas	Total	Pilot	Gas	Total
F-M 7	100	553	8203	8,756	0.31	1.72	2.03
F-M 7	75	641	8484	9,125	0.35	1.77	2.12
F-M 7	50	943	9508	10,451	0.52	1.99	2.51
F-M 8	100	536	8184	8,720	0.30	1.71	2.01
F-M 8	75	656	8417	9,073	0.36	1.76	2.12
F-M 8	50	969	9142	10,111	0.50	1.91	2.41
C-B 1	100	807	8276	9,083	0.46	1.73	2.19
C-B 1	75	1091	8131	9,222	0.62	1.70	2.32
C-B 1	50	1545	8369	9,914	0.88	1.75	2.63

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CHAPTER 24

GAS TURBINES

24-1. Basic Principles. The basic arrangements of gas-turbine engines were reviewed in Secs. 3-10 to 3-15. Now let us study the basic elements of a gas-turbine plant. Figure 24-1 shows the essential components of a single-shaft simple open-cycle gas turbine. It has a *gas turbine* on the right coupled to a rotary-type air *compressor*. The gas turbine drives the compressor and any load, such as a generator, fan, or pump, connected at the load coupling.

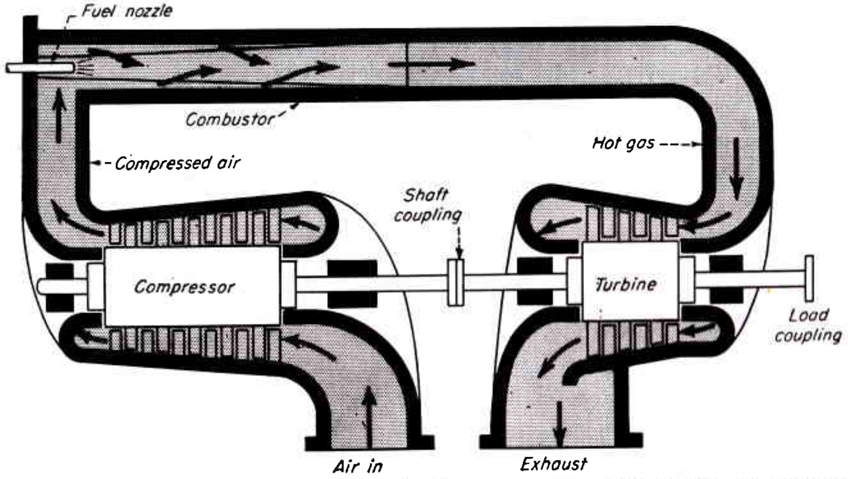


FIG. 24-1. Simple-cycle gas-turbine engine has compressor take in air, pressurize it, and discharge it to combustor. Heated gas-air mixture leaves combustor, enters turbine, and does work on blades mounted on rotor. Rotor drives both compressor and load connected to coupling. Heated exhaust gas discharges from turbine outlet up stack.

Atmospheric air enters the compressor inlet from below. Moving blades mounted on the rotor force the air between the stationary blades to raise its pressure and temperature. The pressurized air leaves the compressor to enter the *combustor*. Part of the air enters the *combustor liner*, or *basket*, to mix and burn with the fuel. The remainder of the air enters the combustor liner through openings farther downstream to mix with and cool the combustion products.

The 3000-F combustion gases must be cooled by the excess air to a level that will not destroy the turbine first-row buckets by exceeding their allowable stresses, which are limited by high temperatures. Gas temperature entering the turbine ranges from 1200 to 1500 F.

As the hot pressurized gas expands through the turbine stages, it develops the motive force for turning the turbine rotor. The same principles work in the gas-turbine stages as in the steam-turbine stages (Secs. 12-1 to 12-6). As the gas leaves the turbine exhaust, its pressure has dropped to atmospheric and its temperature will be in the range of 900 to 1100 F.

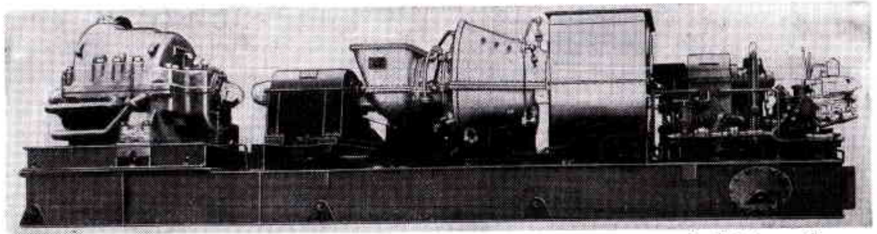


FIG. 24-2. Simple-cycle 3,000-hp gas turbine mounts on common bedplate with gas compressor it drives as load. Left to right, gas compressor, reduction gear, air-compressor inlet, compressor casing, combustor casing (conical section), gas turbine inside exhaust hood, front bearing and reduction gear, starting motor, control and lubricating gear. (*Courtesy of Westinghouse Electric Corporation.*)

24-2. Small Gas Turbine. Figure 24-2 shows a 3,000-hp gas-turbine engine mounted on a common bedplate with its load, a centrifugal gas compressor. Instead of the compressor, this unit could also drive a 2,250-kw a-c generator. From left to right we see the load (gas compressor), reducing gear, air-compressor inlet, air-compressor casing, combustor casing (cone-shaped section), gas-turbine inside exhaust hood, starting-gear casing, and lubricating and control gear. This unit is completely assembled: on arrival at the operating site the inlet and outlet of the gas compressor, the inlet of the air compressor, the fuel-supply line, and the turbine exhaust need only be connected and the unit will be ready to run.

Figure 24-3 shows a cross section through the compressor, combustor, and turbine. The shaft assembly is carried by two bearings, one inside the compressor inlet, left, and the other within the turbine exhaust hood. A segmental type of thrust bearing to the left of the compressor journal bearing takes up the unbalanced axial thrust of the compressor-turbine combination. The reduction-gear case rigidly supports the compressor case, which supports the compressor bearing. Vertical trunnion supports carry the turbine end and bend to allow for thermal expansion. An oil pump pressure-lubricates all bearings of the unit.

Air enters the compressor inlet from above and turns horizontally to the right to flow through the 14-stage axial-flow compressor. At full-load speed of 8,500 rpm the compressor raises 50 lb per sec of air through a pressure ratio of 5:1. The compressor blades have air-foil sections similar to that used in aircraft propellers. The moving blades accelerate the air flow by increasing its kinetic energy; the following stationary blades act as diffusers to slow the airspeed, with concurrent pressure rise.

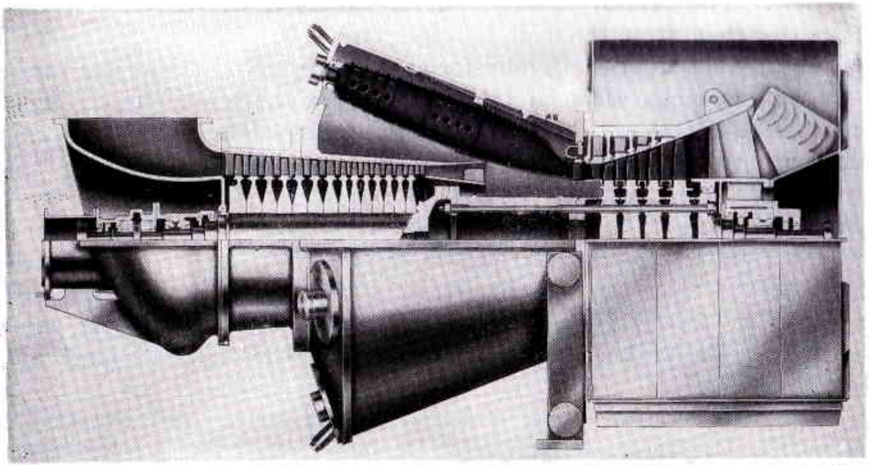


FIG. 24-3. Simple-cycle 3,000-hp gas-turbine engine. Left to right, compressor inlet, 14-stage axial-flow compressor, one of six combustor baskets in combustor casing (fuel nozzle at left), 4-stage gas turbine, turning vanes for turbine exhaust, exhaust hood. Entire rotor assembly carried by journal bearing inside compressor inlet and journal bearing inside turbine exhaust hood. Thrust bearing to left of journal bearing at compressor. Torque tube couples compressor and built-up turbine rotors. (*Courtesy of Westinghouse Electric Corporation.*)

From the last stage of the compressor the air flows through a diffusing section and then makes about a 150° turn into the combustor casing. Some of the air enters the fuel-nozzle end (left end) of the six combustors to mix with the gas or atomized-oil fuel and burn. The remainder of the air enters the combustor baskets through the holes and annular openings to mix with and cool the combustion products. The combustor baskets receive the air so that it blankets the basket metal and protects it from the hot flame; this is called *film cooling.*

The mixed air and combustion products leave the right ends of the six combustor baskets to enter the four-stage reaction turbine at about 1400 F. From the last stage the turbine exhaust gas leaves with a temperature of the order of 900 to 1000 F to enter the guide vanes and be turned upward with minimum turbulence. The gases at slightly more than atmospheric pressure leave through an exhaust stack from the exhaust hood.

This design eliminates all high-pressure air seals and bearings adjacent to hot-metal parts of the turbine and combustors. Cooling air from the compressor exhaust enters the torque tube (hollow shaft) through openings (not shown) and flows through it to the turbine disks. This air divides so that part flows over the faces of each disk to meet the hot gas flow. This film of air cools the turbine disks and the blade roots, insulating them from the main hot-gas flow.

Unheated air from the combustor casing also flows through the space between the turbine casing and the stationary blading diaphragms. This cools the diaphragms, diaphragm-retaining rings, and the cylinder to allow using normal ferritic materials. The entire casing is surrounded by the turbine exhaust hood to minimize thermal stresses from temperature differences.

24-3. Controls and Auxiliaries. Gas-turbine engines need additional equipment to serve the main components; these include: starting motor or engine, motor-driven auxiliary lube pump, starting step-up gear, fuel-control system, oil coolers and filters, inlet and exhaust silencers, and control panels.

The starting motor or engine drives the gas turbine and compressor through a clutch and step-up gear. During starting, stored compressed air expands the clutch member until it grips a clutch drum on the step-up gear. The starting motor then accelerates the gas turbine and compressor to about 600 rpm for the unit in Fig. 24-2. Holding this speed for 5 min assures purging the entire air-gas flow system of any unburned fuel. After accelerating the speed to about 4,000 rpm, fuel enters the combustor baskets and an electric spark in alternate baskets ignites the fuel. Cross tubing between baskets assures fuel ignition in all. The starting motor continues accelerating the turbine to about 5,800 rpm, when it shuts down and the clutch automatically disconnects. Feeding in additional fuel brings the turbine up to rated speed.

The starting step-up gear also acts as the reduction-gear drive for the lubricating-oil-pump-governor impeller and overspeed trip. This engine-driven lubricating-oil pump supplies high-pressure oil to the hydraulic control system and low-pressure lubricating oil for the gas turbine, gears, and driven apparatus. A separate motor-driven pump usually acts as stand-by if the main pump should fail.

For generator drive the fuel feed is made responsive to the speed governor. A drop in speed (increasing load) opens the fuel valves to restore speed to normal. A rise in speed (dropping load) closes the fuel valves to lower the rate of fuel feed and restore speed.

Thermal switches at the turbine exhaust act on the fuel controls to limit exhaust-gas temperature to a preset maximum. Lubricating failure also shuts down the unit automatically.

24-4. Large Gas Turbine. Figure 24-4 shows a 21,800-kw single-shaft simple open-cycle gas turbine. It has a 15-stage axial-flow compressor that compresses about 340 lb per sec of atmospheric air from 15 psia to about 90 psia. The air flow divides equally through 16 combustors arranged in an annular casing about the main shaft. The natural-gas or oil fuel raises the air temperature from about 500 F to 1450 F, using about 20 per cent of the oxygen in the air. In the 2-stage impulse turbine the hot gas drops to about 850 F and 15 psia before exhausting through the hood to the stack.

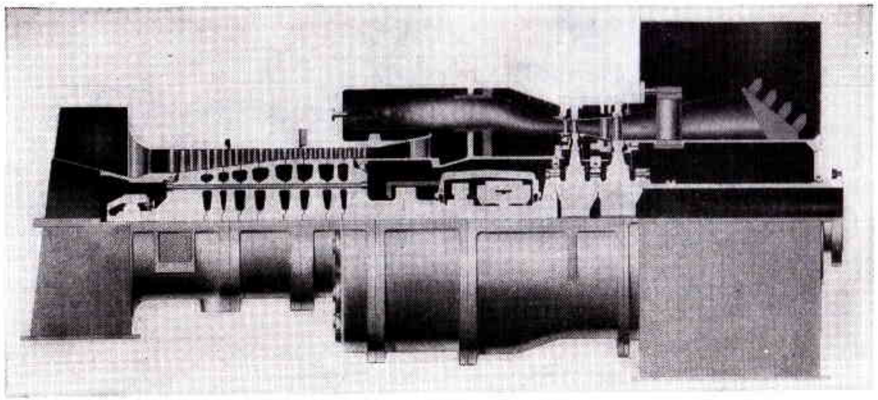


FIG. 24-4. Simple-cycle single-shaft 21,800-kw gas-turbine engine. Left to right, compressor inlet, 15-stage axial-flow compressor, 1 of 16 combustor liners, 2-stage gas turbine, diffusing annulus, turning vanes, turbine exhaust hood. Two journal bearings carry rotor assembly with torque tube carrying load coupling at right. (*Courtesy of General Electric Company.*)

The turbine and compressor turn at 3,600 rpm and couple directly to an a-c generator. About two-thirds of the mechanical power developed by the turbine drives the compressor. This unit also bleeds air from the compressor to cool the turbine disks and blades.

24-5. Regenerative Gas Turbine. Figure 24-5 shows the 3,500-kw regenerative gas-turbine unit of the Texas Power & Light Co. This uses the cycle described in Sec. 3-11; it improves the thermal efficiency by allowing the compressed air to recover some of the energy in the turbine exhaust gas. This preheating reduces the amount of fuel that must be burned to bring the gas up to rated turbine-inlet temperature.

Figure 24-5 shows the layout of the unit with its regenerator, which stands outdoors. All connections between the turbine-compressor unit and the shell-and-tube regenerator run under the operating floor. The

The exhaust duct and stack must also be sized to minimize pressure drop because this loss raises the turbine-exhaust pressure and reduces turbine capacity and efficiency. Exhaust gas from the stack must not be allowed to recirculate to the compressor intake; this can be minimized by increasing stack-gas velocity at the expense of increased pressure loss or increasing stack height, which also raises pressure loss. If there is a prevailing wind, the stack should be placed downwind of the compressor intake. The designer must strike an economic balance between capacity loss and increased investment in larger ducting.

Ducts should be supported from the floor to reduce vibration. They should be stiff enough to resist vibration caused by air and gas flows. Their connection to the turbine and compressor must not put excessive twisting forces on the flanges. Expansion joints in ducts allow for dimensional changes caused by temperature variation.

On inlet ducts acoustical insulation lessens noise from vibration. Turning vanes in elbows reduce pressure loss from vortices in the flowing gas. Manholes give means of cleaning and inspecting the duct interior. Expansion joints and gaskets are usually of synthetic rubber.

Exhaust ducts must withstand temperatures of about 900 F from simple-cycle turbines and about 600 F from regenerative turbines. A 50-ft duct may expand as much as 3 in. from cold and should be spring-mounted to maintain alignment. Expansion joints may be made of glass cloth backed with rock wool for simple-cycle units. For regenerative units or a unit exhausting into a waste-heat boiler metal expansion joints are used. Rock wool with an outer plastic coating usually serves as heat insulation on ducts.

Filters of various types are used on air compressor inlets. The viscous types use a filtering material dipped in oil that catches air-borne particles as they pass through. The dry-type filters use glass fibers as the trapping agent. Either type may be made automatic. The viscous type has an endless curtain of filter material passing through an oil bath for cleaning. The dry type is a long curtain that feeds from a clean-supply roll and takes up on a discard spool. Designers recommend sizing filters to limit air velocity through them to 400 ft per min. This gives a pressure drop of 0.5 to 0.75 in. wg. Dry-type filters have an efficiency of about 85 per cent, and the viscous type of about 92 per cent. Filters must be housed and louvered for weather protection.

Silencers attenuate noise projected from air intakes and turbine outlet to the plant surroundings. Inlet silencers consist of acoustic baffles of sound-absorbing material. Air velocities through them should not exceed 75 ft per sec. Exhaust silencers on simple-cycle units have straight-through flow past an X formed by acoustic panels which consist

of fibrous material enclosed in perforated steel plates. Exhaust-gas velocity runs about 200 ft per sec. Regenerative units and those exhausting to heat-recovery boilers usually do not need silencers.

24-7. Cooling. Most gas turbines use air from the compressor to cool internal parts exposed to maximum temperatures. Water usually proves most convenient for cooling lubricating oil and the electric generator's hydrogen. About 500 to 800 gpm at 95 F will be needed, depending on unit rating.

Figure 3-17 shows the improved turbine performance with cooler compressor-intake air. Air may be cooled by evaporating water in direct contact with it in an *evaporative cooler* or *air washer*. The cooler sprays water into the air after it has passed the filter; mist extractor pads at the cooler outlet prevent moisture carry-over. Coolers reduce inlet temperatures to about 70 or 80 per cent of ambient with a concurrent rise in humidity. Pressure drop through a cooler is about 0.4 in. wg. Make-up water for the cooler ranges from about 3 to 10 gpm.

24-8. Gas-turbine Fuels. Present-day gas-turbine units burn natural gas, blast-furnace gas, distillate oils, and Bunker C oils (residual oil). Development work of many years has striven to design pulverizers, coal pumps, and low-pressure-loss ash separators that will enable a gas turbine to burn coal. The primary problem centers on reducing fly ash in the gas leaving the combustor to a level that will not erode the turbine blading. While encouraging progress has been made, at this writing no commercially successful system has been developed.

Gas pressure must be raised to about 150 psig to ensure proper injection into the combustors. With a pressure ratio of 6:1 combustor pressure runs about 90 psia. Gas pressure must be higher than this to establish good combustion conditions.

Storage and day tanks, transfer pumps, connecting piping, and injection pumps must be used to burn oil. Storage tanks usually hold at least a 2 weeks' supply. Gas turbines can burn either gas or oil in the same unit. Fuels may be transferred under load, burning both while changing over. Such units must have parallel systems for each fuel, but the nozzles in the combustor can burn either or both together.

Mechanical atomizing nozzles may be used for the oils, but air atomizing nozzles are recommended for units making long runs.

Residual oils usually contain sodium, vanadium, and calcium as part of the ash constituent. Sodium corrodes hot metals and builds up hard deposits that choke gas passages in the blading. Vanadium corrodes hot-metal blading at a high rate. To burn residual oils successfully, they must meet the following specifications:

1. The sodium content in ash must be no greater than 30 per cent of

the vanadium content. The maximum sodium content should not exceed 10 ppm, and 5 ppm is preferable. With less than 5 ppm sodium, the ratio of sodium to vanadium is not critical.

2. The magnesium in ash must be at least three times the vanadium. This is not critical when vanadium is less than 2 ppm.

3. Calcium should be 10 ppm or lower. Up to 20 ppm can be tolerated at the expense of higher turbine-nozzle plugging.

4. Lead should not exceed 5 ppm; it nullifies the inhibiting action of magnesium on vanadium.

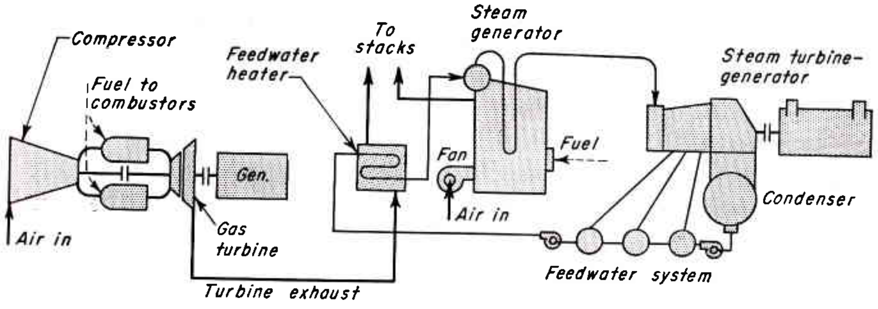
Residual oil may be treated by heating it, then mixing it with 5 per cent of water. Two centrifuges in series receive the mixture to remove the water that takes with it most of the sodium originally in the oil. Just before burning the washed oil, magnesium in the form of epsom salts is mixed with it to inhibit the corrosive action of the vanadium. The treatment costs about 0.3 cent per gallon or about 2 cents per million Btu.

Since residual oils must first be heated to make them fluid, they would plug up fuel-feed lines and combustor nozzles when a gas turbine is shut down and allowed to cool off. To prevent this, distillate oils must first be circulated through the fuel system about 5 to 10 min before shutting down. This flushes all residual oil from the system. Because distillate oils ignite more easily, they are used to bring up a unit from cold before feeding residual-oil fuel to carry load.

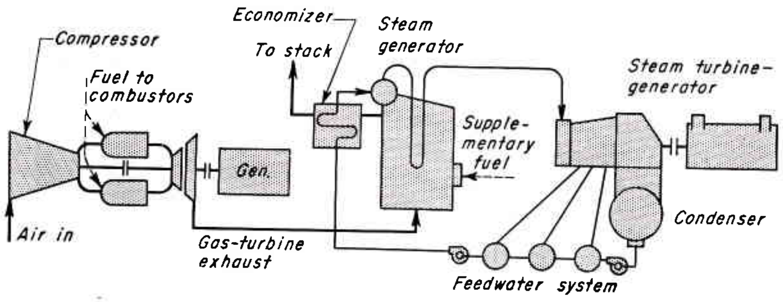
24-9. Combination Gas-turbine Cycles. Gas turbines offer several advantages for different types of service; peak load, emergency stand-by, end-of-transmission-line stand-by, base load, hydro-station stand-by. In some of these services the quick-starting ability makes the gas-turbine plant desirable. In others the lack of cooling water will not hamper a gas turbine while ruling out a steam-turbine plant. Where fuel is relatively cheap oil or gas, a gas-turbine plant often offers better over-all economy for carrying the base load, especially the regenerative, intercooled, and reheat cycles.

The complete versatility of the gas turbine appears when combined with a steam-plant cycle. Gas-turbine exhaust-gas temperatures run about 850 to 1000 F and contain about 16 per cent oxygen compared with 21 per cent in atmospheric air. A simple-cycle gas-turbine plant wastes this energy to atmosphere, while a regenerative gas-turbine plant recovers much of this heat to raise over-all thermal efficiency. But instead we can use the gas-turbine exhaust as a heat source for a steam-plant cycle (Fig. 24-7).

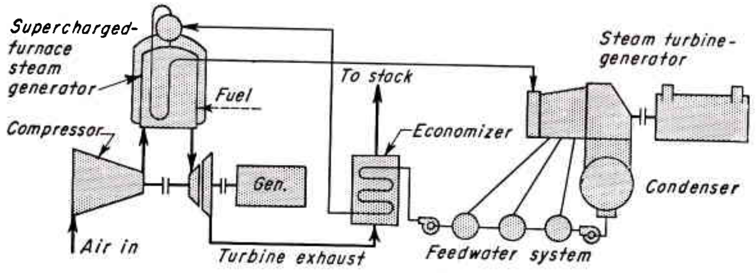
In Fig. 24-7*a* the turbine exhaust passes through a heat exchanger to heat feedwater for a boiler. This application of a gas turbine to a utility system at the Belle Isle Station of the Oklahoma Gas & Electric Co. was the first to be made in the United States. The steam boilers did not



(*a*) Feedwater heating



(*b*) Heat recovery boiler



(*c*) Supercharged-furnace boiler

FIG. 24-7. Combination gas-turbine–steam-turbine cycles offer high-efficiency plants which salvage a large part of turbine exhaust enthalpy. (*a*) Gas-turbine exhaust heats feedwater in conjunction with or alternate to bleed-steam heaters; (*b*) gas-turbine exhaust functions as combustion air for steam boiler or gives up its enthalpy in waste-heat boiler; (*c*) boiler with supercharged furnace acts as combustor for gas-turbine engine, and economizer salvages part of turbine exhaust enthalpy.

have enough capacity to develop the full capability of the turbines. By installing two gas turbines and using the exhaust to heat feedwater, the bleed-steam heaters fed by the turbine could be shut down. This made the full steam-turbine throttle flow expand to the condenser and generate maximum possible kilowatts. In addition there was a slight increase in over-all plant thermal efficiency, because the gas-inlet temperature to the turbine was 1450 F, compared with the conventional 900 F at the steam turbine.

TABLE 24-1. COMPARISON OF CONVENTIONAL AND SUPERCHARGED BOILER CYCLES
Steam cycle: 1,250 psig, 950 F, 1.5 in. Hg, oil-fired

	Conventional steam cycle	Combined cycles	Combined cycle improvement, %
Net output, kw	44,570	44,570	
Steam turbine, kw	46,450	40,160	
Gas turbine, kw	0	5,700	
Steam generation, lb per hr	382,000	319,400	
Condenser flow, lb per hr	275,190	248,300	
Condenser heat rejection, Btu per lb	921.2	919.7	
Stack-gas flow, lb per hr	401,500	377,000	
Stack temperature, °F	300	300	
Losses:			
Station auxiliaries, kw	1,880	1,290	0.4
Mechanical and electrical loss, kw	1,030	1,290	−0.2
Boiler radiation, 10^6 Btu per hr	9.4	4.3	1.1
Condenser loss, 10^6 Btu per hr	253.5	228.4	5.4
Dry stack loss, 10^6 Btu per hr	22.3	21	0.3
Stack moisture loss, 10^6 Btu per hr	21.7	20	0.4
Total over-all plant improvement, %	....	....	7.4
Fuel fired, lb per hr	25,480	23,600	
Total heat release, 10^6 Btu per hr	467.6	433.1	7.4

Figure 24-7*b* shows a combined cycle using the gas-turbine exhaust as preheated "air" to the steam generator. The hot 16-per-cent-O_2 gas contains enough oxygen to support combustion. By making the steam generator large enough and feeding supplementary fuel to its furnace the steam turbine will generate about five times the kilowatts produced by the gas turbine. Improvement of about 5 per cent in over-all plant heat rate can be realized.

Without supplementary fuel firing the boiler could be a waste-heat boiler and with a smaller steam turbine simply improve the efficiency and expand the capacity of the over-all plant. This cycle has been used in several industrial plants and at least one central station.

Considerable study has been made of the cycle in Fig. 24-7*c*. Here the combustors of the gas-turbine unit are replaced by a steam generator having a supercharged furnace, and the gas-turbine exhaust heats the feedwater before it enters the boiler. Because of the 70 psig furnace pressure much less surface is needed to transfer a given amount of heat to the steam and water in the boiler and superheater. Where conventional boilers transfer heat at the rate of about 85,000 Btu per hr-sq ft, the supercharged boiler can work at about 200,000 Btu per hr-sq ft and with development might be stepped up to about 450,000 Btu. This shrinks the over-all size of the steam generator, with marked reduction in investment. The cycle also eliminates forced-draft (f-d) and induced-draft (i-d) fans, with their power consumption.

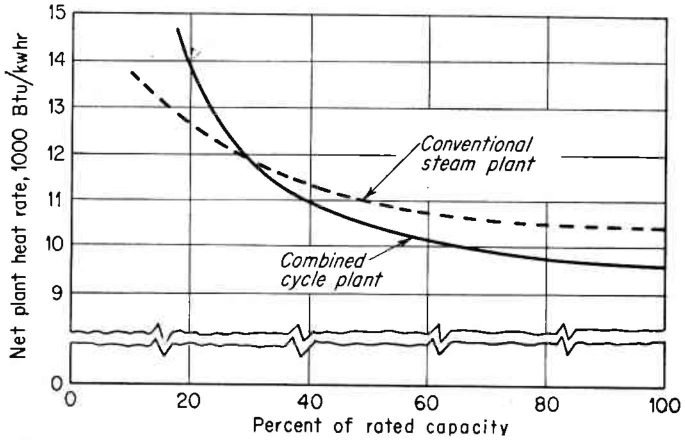


Fig. 24-8. Combined gas-turbine–steam-turbine plant works with superior efficiency compared with conventional steam-turbine plant.

John W. Mann, Jr.,* found the following comparison for a 45-mw plant, as in Table 24-1: Over-all heat-rate improvement for the combined cycle compared with an equal-capacity steam plant was 7.4 per cent. The 1450-F heat addition to the gas turbine lifted cycle efficiency compared with the 950-F all-steam cycle. The table shows that the steam-turbine generates about 85 per cent of the total output. Inlet steam flow, condenser flow, and gas flow at the stack are in about the same ratio. Reduction in the steam flow to the condenser makes the largest saving. The lower stack-gas flow reduces the stack loss in the combined plant. The saving in auxiliary power to boiler fans is slightly offset by the loss in gas-turbine generator reduction gear.

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* John W. Mann, Jr., Thermodynamic Performance and Design of Steam–Gas Turbine Power Plants, *Proceedings of the American Power Conference*, vol. 18, pp. 299–300, 1956.

Other estimates show that the investment for the gas-turbine unit and supercharged boiler roughly equals the cost for a conventional steam generator with its fans which are replaced in this cycle. So it appears that this more efficient cycle can be installed at no more than conventional over-all plant investment. At this writing no plans for an actual plant have been announced.

Figure 24-8 shows how the heat rate of a combined plant varies with load and compares it with an equal-sized steam plant. At about 30 per cent of rated capacity the heat rates of both plants are equal, but for higher outputs the combination plant proves superior. In this study the compressor raises 355,000 lb per hr air through a 6:1 pressure ratio. Turbine inlet temperature is maintained at 1450 F. Fuels at present are limited to gas and oil until some means of removing ash from furnace gas will allow burning coal.

PART IV

HYDRO AND METEOROLOGICAL PLANTS

CHAPTER 25

WATER POWER AND HYDRAULIC STRUCTURES

25-1. Water Power. Rain falling upon the earth's surface has potential energy relative to the oceans toward which it flows. This energy can be converted to shaft work where the water drops through an appreciable vertical distance. Such a site develops power,

$$P = \frac{Qh\eta}{8.8} \qquad (25\text{-}1)$$

where Q = rate of water flow, cfs
h = height of fall or head, ft
η = efficiency of conversion of potential energy to shaft work, fraction

The efficiency of well-designed plants may approach 90 per cent. The equation shows that the power developed depends on *quantity* and *head* of water.

The usual hydraulic plant need be concerned only with the water flow from its immediate point of origin above the plant site to its point of discharge from the site. The energy derived from falling water is a very small process in the gigantic heat-power cycle, technically called the *hydrological*, or *rain-evaporation, cycle.*

The energy input to this cycle, in the form of solar radiation, evaporates vapor from the surfaces of water bodies covering the earth's surface. The vapor molecules absorb this energy in the kinetic form, giving them the ability to rise and do work against the force of gravity. Depending on complex meteorological conditions, the vapor molecules form clouds as they condense and give up energy. Further rejection of energy makes them precipitate in the form of rain, snow, hail, or sleet. The mechanical energy involved in the rising and falling of the vapor and precipitation is many times larger than that available at any earth-bound waterfall. Despite the enormous circulation of energy no practical means of using it has been devised beyond the present-day hydraulic plant.

25-2. Hydrological Cycle. Cooling our atmosphere of air and water vapor makes some of the vapor condense and fall as rain, hail, snow, or sleet. When this precipitation falls on hills and mountains and converges to form streams, we can possibly use it for power generation. No two geographical regions have identical precipitation experiences; this makes each proposed site for a hydraulic plant a unique problem. Usually coastal regions or regions adjacent to large bodies of water have greater amounts of rainfall than inland regions. Precipitation generally increases with altitude on the coastal side of mountain ranges up to the limit of cloud elevation.

Intensity of rainfall, season, and topography largely determine the usefulness of rainfall for power purposes. Light falls aid the growth of vegetation but do not contribute to stream flow. When total monthly precipitation concentrates in one or more storms, the *runoff* will increase greatly though vegetation may suffer. Extensive observations show that relatively small proportions of rainfall can be used for power purposes. Distribution or disposal of precipitation may be classified as: (1) direct evaporation, (2) absorption and transpiration by vegetation, (3) seepage and storage, and (4) direct surface runoff, eventually forming rivers.

1. A significant part of the rainfall never reaches the soil but clings to the surface of vegetation and reevaporates to the atmosphere. The major part of precipitation on land areas does reach the soil and quickly reevaporates to the atmosphere. The evaporation from the surfaces of ponds, lakes, and swamps is very large; in fact most of the water so stored usually disappears by evaporation rather than by *seepage* or runoff.

Although the rate of evaporation varies erratically, it follows a general cycle corresponding to the four seasons of the year. Variations depend on changes in atmospheric conditions, such as temperature, humidity, cloudiness, and wind. Evaporation rate depends directly on temperature and humidity; a rise in temperature and drop in humidity increase the evaporation rate. Wind aids evaporation by constantly removing the blanket of vapor molecules from the water surface, provided that the wind is not saturated.

2. The character of the vegetation and the soil in which it grows affect the amount of rainfall it absorbs. Plants absorb all their moisture through their roots and *transpire* it as vapor through their leaves to the atmosphere. However, vegetation shading considerable ground area from the sun reduces the direct evaporation. On the other hand, transpiration increases markedly during the growing season, when sunlit hours are a maximum. Most vegetation lies dormant during the winter, making transpiration negligible.

3. Precipitation absorbed by the soil *seeps*, or *percolates*, into the ground, forming bodies of water called the *water table*, or *ground storage*. The amount of *seepage* depends on the geological character of the surface and subsoil. Cultivated or plowed land encourages seepage provided that no impervious subsoil layer interferes and that the water sinks below

the reach of the lowest roots of the vegetation. Sandy subsoil offers the best condition for seepage; the finer the soil particles, the less favorable are conditions for ground water. Most ground water eventually flows into streams and during droughts is the only source of water to maintain stream flow.

Lakes and swamps store much water, primarily by rising water levels since they have restricted outlets. But most of the water evaporates instead of discharging through the outlet as useful runoff.

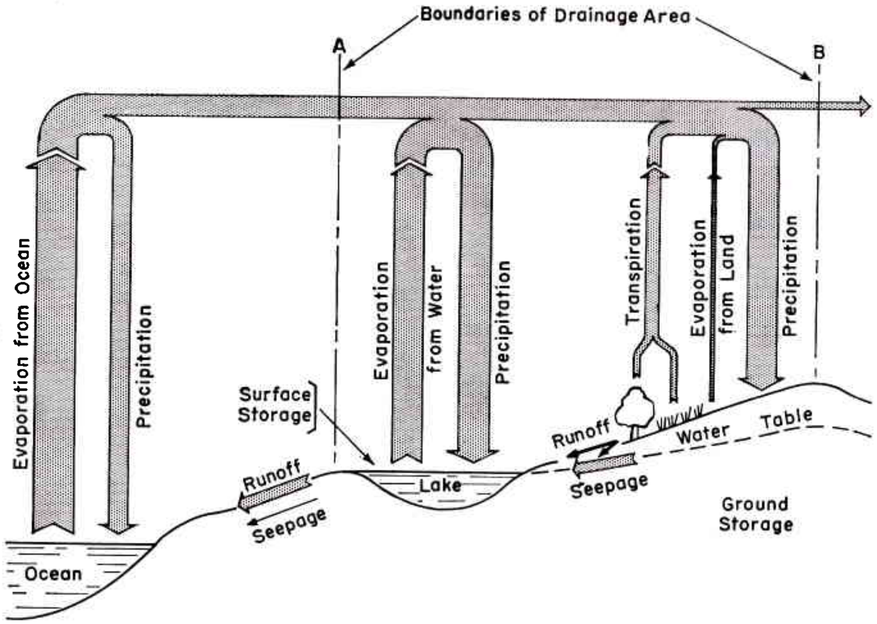


FIG. 25-1. Graphic portrayal of the water cycle. (*Courtesy of V. A. Thiemann.*)

4. Water not distributed in one of the three ways described flows over the ground surface as direct runoff to form brooks and rivers. The suitability of a river for power generation depends on the magnitude and time distribution of flow and the head made available by the surrounding topography.

The amount of runoff from a given rainfall depends on the nature of the precipitation; short, hard showers may produce relatively little runoff, whereas rains lasting a longer period usually produce a greater percentage of runoff. The soil tends to become saturated and lowers the rate of seepage and the humid atmosphere slows evaporation.

During the winter, with frozen ground, rainfall may appear almost completely as runoff. During this season snow and ice store much of the precipitation to be released during warmer weather as runoff. Melting snow and ice augmented by current rainfall cause the familiar spring freshets or floods.

The water equation summarizes the disposal over a definite area of the rain falling during a given period, as in Fig. 25-1:

$$\text{Runoff} + \text{seepage} = \text{precipitation} - \text{evaporation} - \text{transpiration} \pm (\text{change in storage}) \tag{25-2}$$

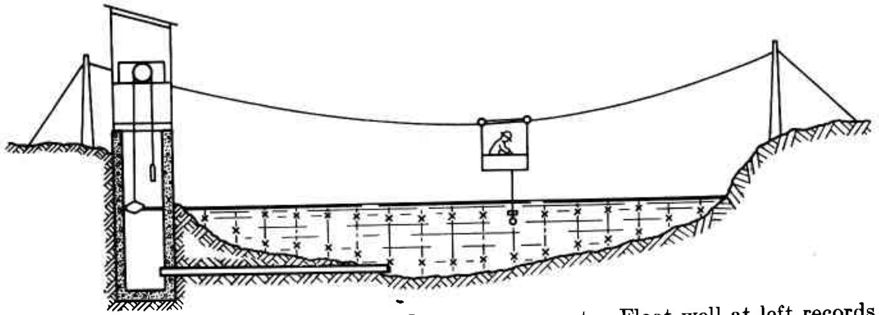


Fig. 25-2. Gauging station for stream-flow measurement. Float well at left records water level; river flow is being measured by current meter from cable car. (*Courtesy of V. A. Thiemann.*)

The water leaving at boundary A is the difference between the vapor leaving at boundary B and that entering at A, with allowance for accumulation or diminution of storage.

25-3. Measuring Stream Flow. To evaluate the available energy in a given river we must know the quantity of water flowing and its variation with time over a long period of years. Direct measurement by stream gauging at a given site for a long enough time is the only precise method of evaluation.

The water-flow volume may be measured by providing or selecting a channel of fixed cross section and at regular intervals measuring the water velocity at enough points in the cross section for different water levels, or *stages.*

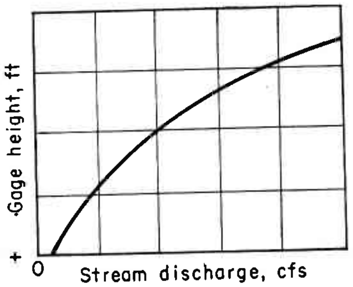


Fig. 25-3. Typical curve relating water level at gauging station and stream flow.

Figure 25-2 shows a typical stream-gauging station; by integrating the velocities over the cross section for each stage the total flow for each stage can be calculated (Fig. 25-3). With this curve we need only a continuous record of the river stage, provided that the gauging-station cross section remains fixed.

The U.S. Geological Survey and cooperating authorities established gauging stations on most important rivers about 1906 and have increased the number of sites since then. Many sites have automatic recorders; at others observers make records manually several times a day or week depending on the variation and rapidity of flow.

The flow of a river may be given in terms of second-feet average discharge per day, per month, and per year. This discharge multiplied by the head gives the energy available for conversion. Alternate methods of expressing flow or runoff are inches of water on the drainage area feeding the river site for a stated period, or acre-feet of water per unit of time.

Besides the annual total and average rates of flow the maximum and minimum rates must be known. Maximum flood flows can be estimated only on the basis of probabilities, since rainfall acts erratically. By using available data on runoff the mathematical theory of probabilities can predict that, on the average, floods of given magnitudes will occur once in so many years. The greater the flood, the longer the span of years intervening.

The maximum flood flow governs the size of the headworks and dam to be built. Their cost must be balanced against the damage that greater floods might cause. If failure of a dam would cause loss of life below the site, the dam would probably be designed for a flood occurring once in 1,000 years or even 10,000 years. A dam on a remote river might be designed to hold floods occurring once in 10 or 20 years, the recurring costs of repairing damages caused by greater than the design flood being cheaper in the long run than installing a larger, costlier dam initially. In most cases, if a dam were built to resist the maximum possible flood of all time, the high cost would make the venture unattractive economically.

Average flows must be known to figure the maximum marketable energy and the revenue the projected plant may be expected to yield. The capacity of the plant and the need for storage will also be fixed by the average flow.

The minimum flow and that part which can be stored fix the amount of primary power that will be available. That part of the capacity of a hydraulic plant available at *all* times when energy is needed is classed as *primary power.*

The additional output available at high-water flows, called *secondary power,* sells at a much cheaper rate than primary power because it can be revoked at will of the operating company. When the market for primary power is larger than the hydraulic plant can supply, the difference can be supplied by a steam plant or diesel plant or over an interconnection with another utility system.

Sometimes the lack of direct runoff data makes it necessary to estimate stream flow. With a few years of measurement available a correlation may be established with a neighboring stream. Due account must be taken of differences in rainfall, topography, and geology.

When no gauging data are available, the estimates must be based on available rainfall data. These are on record for many years in most parts of the United States, longer than existing stream-flow data. Several equations have been proposed to correlate rainfall with stream flow. No direct reliance should be placed on such correlations without a thorough analysis of other controlling factors.

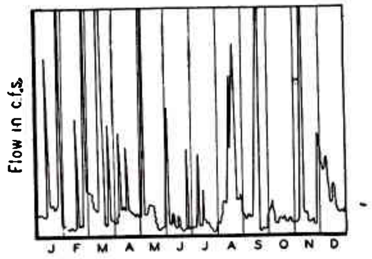


FIG. 25-4. Annual hydrograph of a river with flashy characteristics.

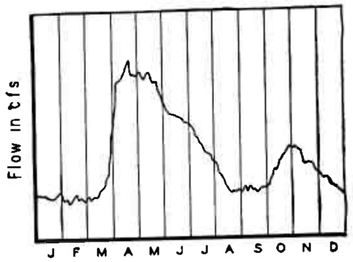


FIG. 25-5. Annual hydrograph of a river with exceptionally steady flow.

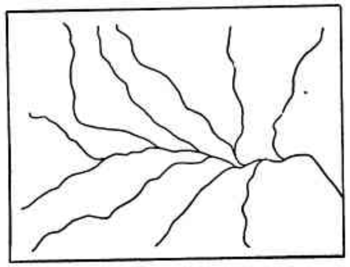


FIG. 25-6. Map of river with fan-shaped tributary system conducive to flashy flow.

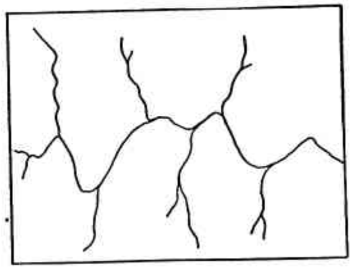
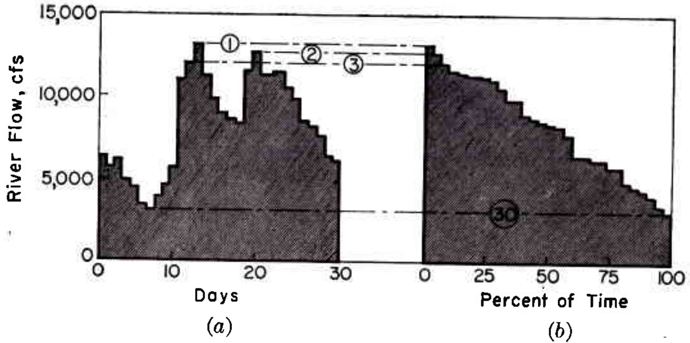


FIG. 25-7. Map of river with extended tributary system conducive to steady flow.

25-4. Drainage-area Characteristics. The variation of stream flow in a given site depends on the geographical, geological, and topographical features of the drainage area feeding the river, as well as the magnitude of the area rainfall. *Hydrographs* show the variation of river flow with time; Fig. 25-4 shows the behavior of a flashy stream and Fig. 25-5 the variation in a stream with relatively steady flow. Steep slopes on the drainage area, impervious soil surfaces, impervious subsurface strata, sparse vegetation, and a fan-shaped distribution of tributary streams merging near a common point (Fig. 25-6)—all contribute to flashy behavior in a river. Gentle slopes, soils and rock formations conducive to seepage, ample vegetation, and tributary streams that enter at widely separated points (Fig. 25-7)—all contribute to relatively steady flow in a river.

The effects of deforestation find no agreement among authorities, but most hydrologists feel that forests do not prevent floods. The underbrush in heavy forest growth undoubtedly lessens the severity of floods by retarding the rate of flow, but the principal benefit lies in reducing land erosion and silt loadings of river waters.

Runoff may also be plotted as *flow-duration curves* (Fig. 25-8). These show the time that a stream-flow rate was equaled or exceeded in any period. In Fig. 25-8 the highest daily flow is shown at the highest level at 1 and the lowest at the lowest level at 30.



(*a*) (*b*)

FIG. 25-8. Flow-duration curve (*b*) devised from hydrograph (*a*) type of plotting. (*Courtesy of V. A. Thiemann.*)

25-5. Storage and Pondage. Hydraulic plants may be classified according to the amount of water-flow regulation they can exert. A run-of-river plant has no control of river flow and must use the water as it comes. During high water-flow rates and at times of low load considerable water is wasted over the dam spillways.

Storage plants vary in degree. Plants with facilities for storing water at their sites often cannot store as much as plants having one or more reservoirs upstream. Water behind the dam at the plant is *pondage*. Water held in upstream reservoirs is *storage*.

The amount of regulation obtained with pondage usually involves storing water during low loads (usually during early morning hours and Sundays) to aid carrying peak loads during the week. The water that would go over the dam spillway unused during low loads (as in run-of-river plants) can be released and added to normal river flow to supply peak loads, usually of a few hours' duration.

The headwater elevation fluctuates with this type of operation: rising when storing water (load less than energy in river flow); falling when drawing water (load greater than energy in river flow); and remaining constant when load equals energy in river flow. For fluctuating loads pondage increases the maximum capacity that a plant can carry.

Plants with reservoirs upstream can store the excess water of spring floods for release during the summer to supplement the low rates of flow during this dry season. Reservoir water elevation will generally be lowest during the year at the end of the summer.

Pondage increases the capacity of a river over a short time, such as a week. Storage, however, increases the capacity of a river over an extended period such as 6 months to as much as 2 years.

25-6. Dams. Dams function in two ways: (1) creating a head and (2) impounding water for storage. Either or both may be performed by one dam. Of course, to create a head, some water must be impounded, but small amounts are not considered useful storage.

The designer uses stability and tightness as the major criteria in building dams. Dams must be designed to resist failure by upsetting, sliding, and rupturing. This may be done by sheer weight, as in solid dams, or by anchoring, as in arch dams. Tightness depends on: (1) imperviousness of the structural materials, (2) adequate joint design, and (3) sound geological formations of the terrain surrounding the dam on all sides.

Large seepage through subterranean passages may make a dam ineffective. When incomplete geological surveys do not reveal such underground leaks, expensive plugging repairs must be made. Experience shows that only a few borings at a site may appear to make it favorable for a dam, but more extensive borings may reveal serious subsurface faults that make it wise to abandon the site in favor of one needing perhaps a larger dam and having less storage capacity.

Dams may be classified according to their structural materials as: *timber*, *earth*, *rock-filled*, and *masonry*. Some dams combine two of these, such as a solid masonry section with a spillway in the center flanked by earth sections on both sides. The type selected depends on foundation conditions and economics.

Few timber dams are built today. They were used for sites remote from roadways and where timber was plentiful. These dams were built as high as 40 ft, with averages around 20 ft. Timber dams need constant inspection to keep them in good repair.

Earth dams are built to heights of 200 ft. They contain the maximum cubic content of material and have the marked advantage of not needing an impervious rock foundation. The dam cross section varies with its height and the nature of the material. The side slopes of the dam must be considerably flatter than the limiting angle of repose of the material when under water. Figure 25-9*a* shows a dam without a core wall. The line of saturation is the height to which water would rise in a pipe driven vertically into the dam.

The saturation line must intersect the dam's base line well within the downstream toe of the dam; otherwise, heavy seepage of the downstream side may make the entire dam fail. To depress the elevation of the saturation line, a core wall or puddle may be placed in the dam section as in Fig. 25-9*b*.

The dam must be high enough so that flood flows will not top it; if they do, the dam will quickly fail from erosion. A protecting coat of rock, concrete, or planking must be laid at the water line to protect the dam from wave erosion. Covering all exposed surfaces with grass or vegetation protects the dam from rainfall erosion.

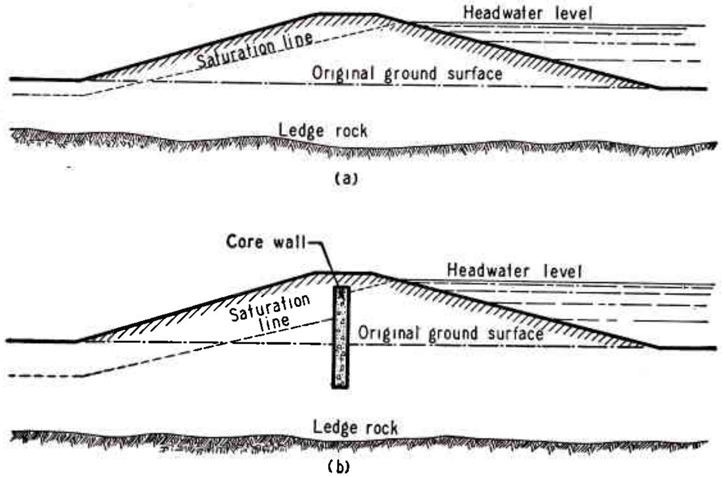


FIG. 25-9. Earth-dam cross sections. (*a*) Without core wall; (*b*) with core wall.

The rock-filled dam, used mostly in the West, consists of rock blasted from nearby borrow pits and either dumped or carefully placed in position. In building a dam between canyon walls, the rock may be blasted from the sides to fall into position. Concrete or timber mats placed on the slopes or concrete core walls limit seepage through the dam. These dams usually settle with time unless carefully built; they will fail if topped by floodwater.

Masonry dams are of three major classes: (1) the *solid gravity dam*, (2) the *buttress dam*, and (3) the *arched dam*. The mass of the dam structure decreases for a given site in the order named. For maximum safety all dams should be built on a solid, impervious rock foundation. When this is not available, the first two types may be built with a concrete mat or apron laid over the pervious earth foundation to an extent that will ensure a stable structure.

Figure 25-10 shows the cross section of a solid gravity dam. This may be made of ashlar masonry, mortar rubble masonry, or concrete with large stones embedded. This dam is solid through its length except for joints in the structure to allow for expansion and contraction with temperature.

In stable dams the resultant of all forces acting on the dam will intersect the base at some point within the middle third of the base line. Then

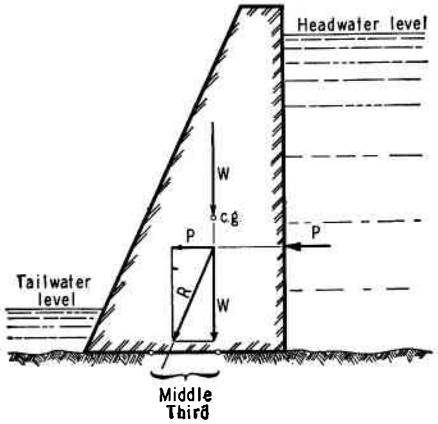


FIG. 25-10. Cross section of solid gravity type of masonry dam. This type may be built as a *nonoverflow* dam or as an overflow type called a *spillway* dam. The latter type has a downstream apron as part of the dam. On this apron the kinetic energy of the water is partly changed to potential energy in the tail water characterized by the *hydraulic jump* as the water enters the tail way.

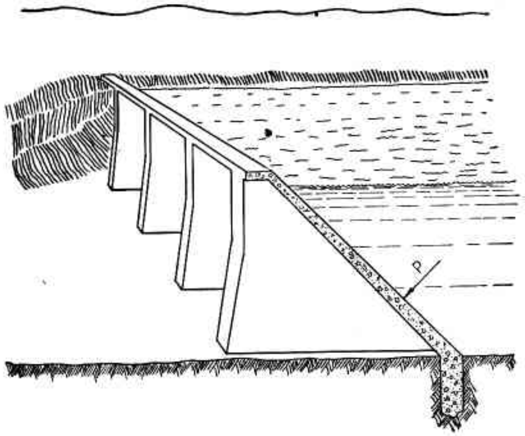


FIG. 25-11. Buttress or hollow gravity type of masonry dam with flat deck.

all points of the base exert downward pressure on the foundation, the criterion of stability. Figure 25-10 shows the water pressure P and the weight of the dam W as the principal forces acting on the dam. R, the resultant of these two components, intersects the base at the required

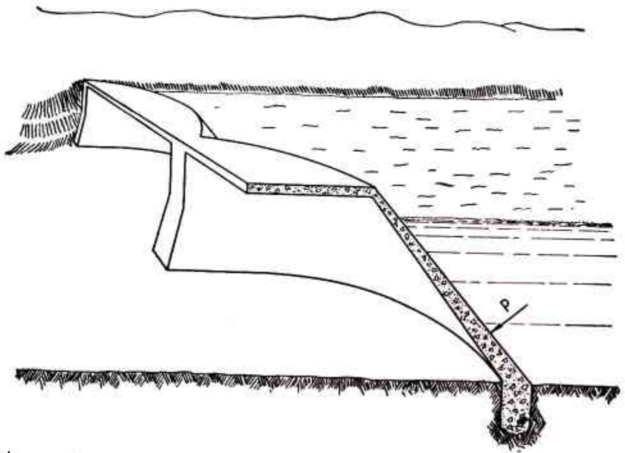


FIG. 25-12. Buttress type of masonry dam with multiple-arch deck.

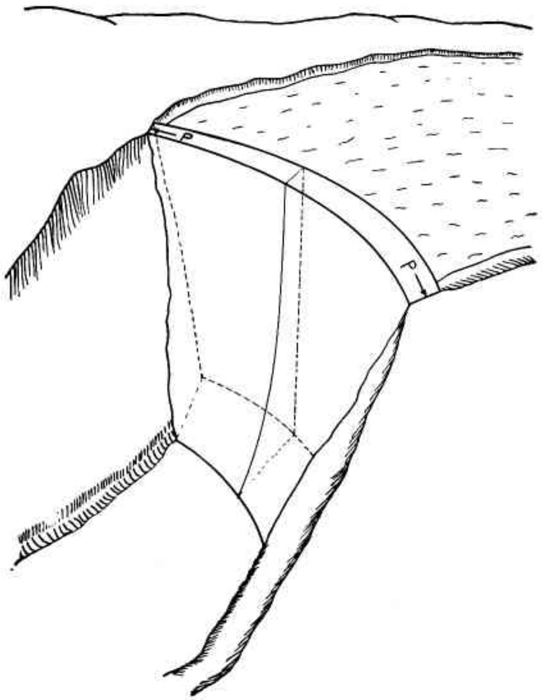


FIG. 25-13. Arch type of masonry dam.

middle third. Friction between the dam and the foundation balances the component that causes sliding.

The hollow gravity dam (Fig. 25-11 or 25-12) may be selected when concrete costs are high. The dam shown in Fig. 25-11 has a flat deck

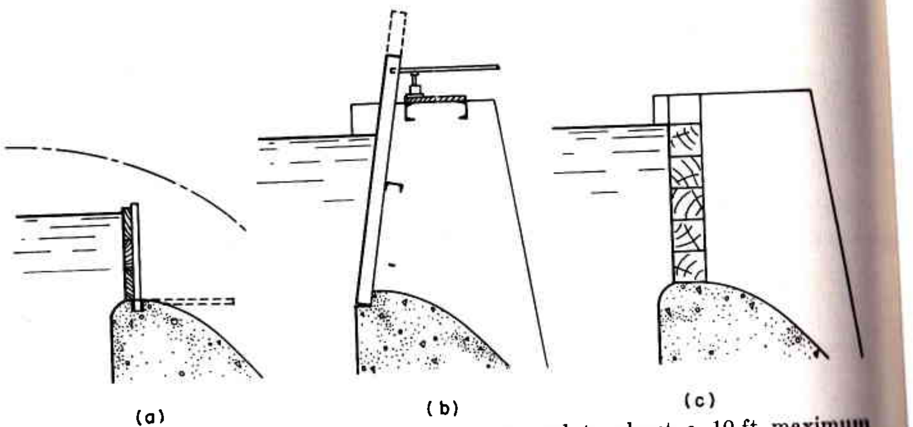


Fig. 25-14. (*a*) Typical flashboard arrangement used to about a 10-ft maximum height; (*b*) timber needles jacked up out of notch let water pass to lower level; (*c*) stop logs laid on top of each other raise water level as needed.

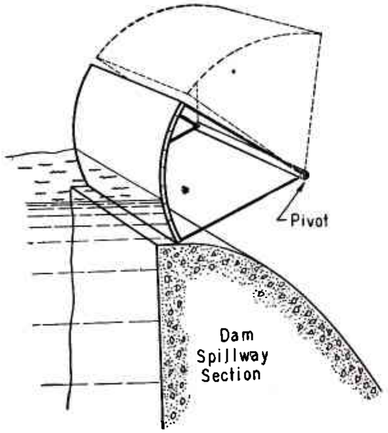


Fig. 25-15. Tainter gate for headwater-level control. Hoist mechanisms are used to set the gate between closed or full-open (dotted) position. Gate is usually of steel, sealed against leakage at side and at bottom when closed, and closes by its own weight.

laid on sloping buttresses. The weight of water *P* on the deck is transmitted as a thrust into the buttresses. The weight of the water and the angular direction of its pressure make up for the lighter weight of this dam in assuring stability.

The multiple-arch type uses the same principles except that the deck arches between buttresses, curving into them tangentially. This design can use wider buttress spacing and save concrete. The additional formwork needed for the hollow gravity dam may offset its economy of materials.

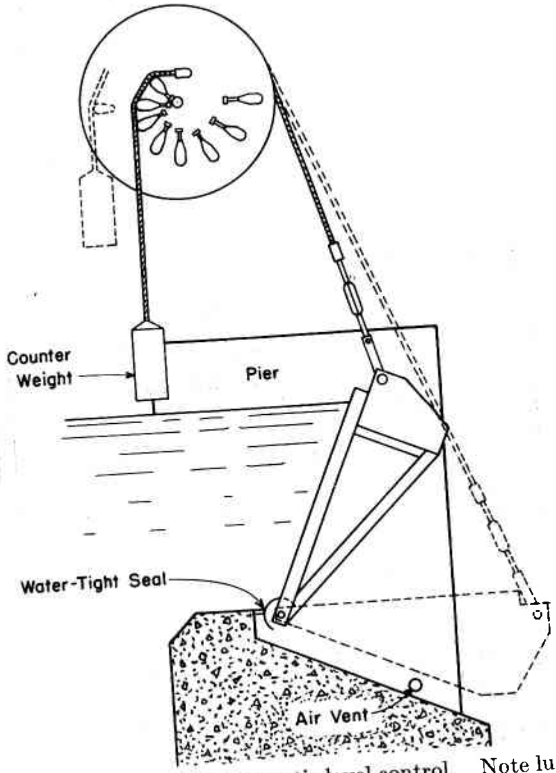


Fig. 25-16. Typical tilting gate for automatic level control. Note lugs to vary torque produced by counterweight. Air vent relieves vacuum formed under open gate. (*Courtesy of V. A. Thiemann.*)

The arch dam (Fig. 25-13) has all forces acting perpendicular to the surface so that they tend to compress the material in the arch. Resisting forces at the feet of the arch balance the compressive forces. This dam proves suitable where the width is narrow compared with the height. The sides of the channel must be composed of rock capable of withstanding the thrust exerted by the feet of the arch. The arch dam requires the least amount of materials.

25-7. Spillways. Dams must be able to pass flood flows that might damage them. This may be done with: (1) spillways, (2) conduits piercing the dam, or (3) tunnels bypassing the dam. A plain spillway section in a dam can spill excess water but has certain disadvantages. To make the maximum head available, the spillway crest should be as high as possible. But the elevation permissible during normal river flows will be excessive during floods because the water will back up and flood upstream land. This may cause damage upstream, and so flowage land or rights must be obtained for the maximum level of water.

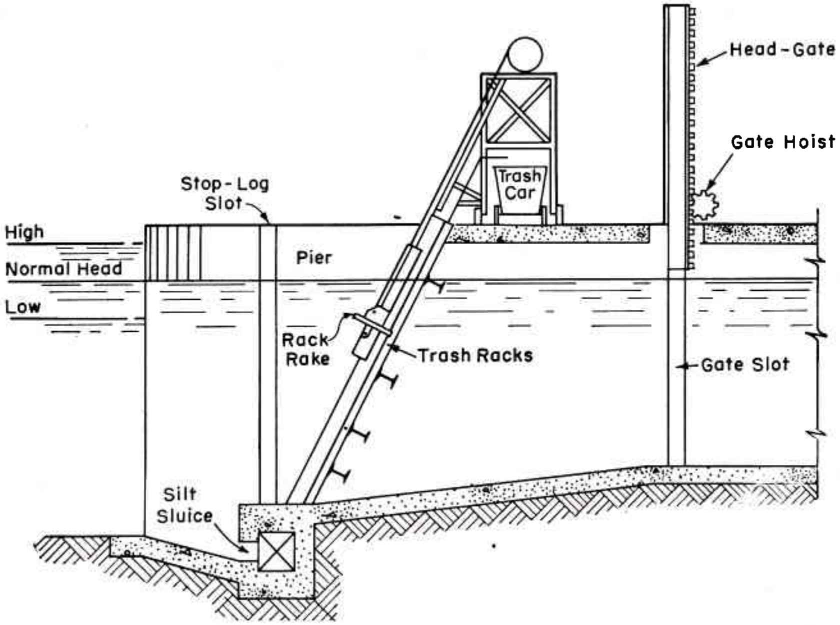


Fig. 25-17. Typical low-pressure intake showing headworks for a hydro plant. (*Courtesy of V. A. Thiemann.*)

For flexibility the head level should be controlled independently of the flow through the turbines. One method uses a superstructure at the crest to hold: (1) flashboards or (2) timber needles or (3) stop logs. Figure 25-14*a* shows a flashboard construction for automatic control of headwater level. Sockets at the top of the spillway support iron rods projecting as high as 10 ft above the crest. Flashboards lie against these pins held by the force of the water. When water begins spilling over the board tops, the additional pressure on the boards makes the pins or rods fail by bending; this releases the boards, allows the water to pass at a higher flow, and lowers the level in the head pond (plant intake).

Superstructures hold timber needles (Fig. 25-14*b*) or stop logs (Fig. 25-14*c*) on top of the dam crest to control water level. Jacks or hoists must remove or add the needles and logs as water height is to be varied.

Modern dams use some type of crest gate: (1) radial, (2) flat, (3) roller, (4) drum, or (5) tilting. Motors controlled by float switches raise and lower the first three types of gates. Drum and tilting gates operate automatically with changes in headwater level. Figure 25-15 outlines the principle of a radial gate, called a Tainter gate. Figure 25-16 shows a tilting gate; as water level rises, the pressure on the gate forces it downward to spill water. As the level drops, the heavier counterweight overbalances the weight of reduced water height flowing over the gate to pull it up and stop water flow.

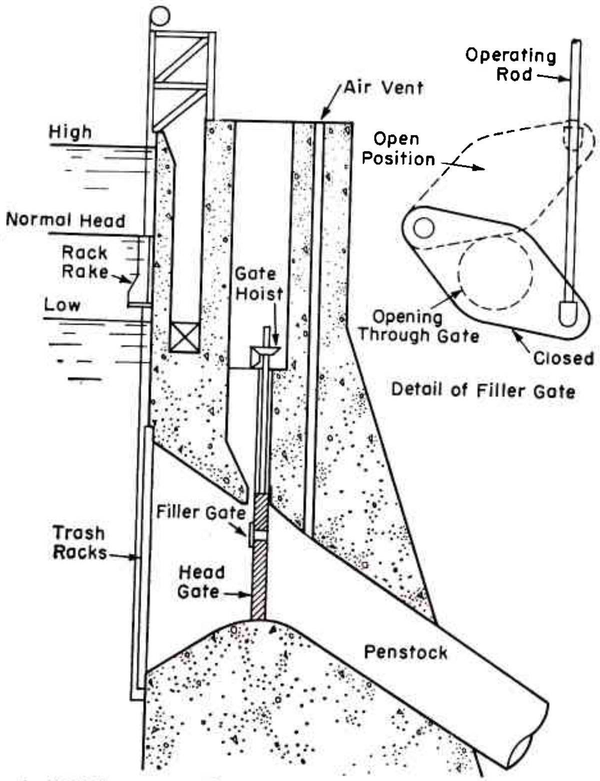


Fig. 25-18. Typical high-pressure-intake headworks. Note that filler gate balances water pressure for opening the gate; air vent prevents penstock vacuum. (*Courtesy of V. A. Thiemann.*)

Water flowing over a spillway acquires considerable kinetic energy, that can do considerable damage downstream of the dam. The spillway must be extended from the dam base by an *apron* section. The water entering the apron at high velocity enters a *hydraulic jump* that absorbs much of the kinetic energy and prevents eroding the river bed. Water spilled from conduit outlets dissipates its energy in turbulence as it drops into downstream sections of the river.

Where dams offer barriers to fish migration, log driving, or navigation, they must have suitable bypasses: fishways in the form of ladders or elevators, chutes for passing logs, and locks for ships.

25-8. Headworks. These are structures at the intakes of conduits, tunnels, and flumes. They have booms, trash racks, sluices for passing debris, and gates or valves for controlling water flow.

Booms divert ice and floating logs from the intake. They consist of logs tied end to end and form a floating chain that directs floating matter to a diversionary bypass chute.

Trash racks remove floating or submerged debris directly at the intake entrance. Gates control the rate of water flow entering. Figure 25-17 shows a typical low-head intake and Fig. 25-18 a high-head intake.

Other high-pressure or high-head headworks use cylinder gates, or rotary, spherical, butterfly, or needle valves for water-flow control. These gates are used not only at headworks but also in penstocks and at the turbine inlets. They also control discharge through sluices and conduits for flood-flow bypasses.

25-9. Conduits. A *headrace* leads water from the headwater to the turbine; a *tailrace* leads water from the turbine outlet to the downstream side of the plant. These may be open-type flumes or canals or closed-type tunnels, pipelines, and penstocks. Many plants use a combination of these waterways. Penstocks handle high-head water and may be made of reinforced concrete, wood staves, or steel plate. They may be buried or aboveground. Penstocks have accessories such as air vents, surge tanks, and pressure regulators to protect the penstock against sudden pressure changes that might burst it or collapse it under sudden load and flow changes.

PROBLEMS

25-1. At a potential hydraulic-plant site the average elevation of the headwater is estimated at 1,812 ft, the tail-water elevation at 1,575 ft. The average annual water flow was determined to be equal to that volume flowing through a rectangular channel 30 ft wide with a depth of 4 ft and an average velocity of 15 ft per sec. Find the annual electrical energy this site can produce with a water turbine efficiency of 85 per cent, a generator efficiency of 95 per cent, and the loss in the headworks equal to 3 per cent of the available head.

25-2. A lake having a surface area of 150 sq miles receives its water as runoff and seepage from the drainage of an area of 2,000 sq miles including its own area. The lake empties through a single outlet. The lake level was 1,800 ft at the beginning of a given month of 720 hr and 1,802 ft at the end. The average rainfall over the area during the month was 4.0 in. If 40 per cent of the precipitation was evaporated and transpired, what was the average flow through the outlet in second-feet during the month?

25-3. The river forming the outlet of the lake in Prob. 25-2 has a rapids starting at the lake which descends 220 ft within a short distance. What energy could have been developed ideally at this site for the month in the foregoing problem?

25-4. A factory is situated at a fall of 60-ft drop in a river. The factory requires a source of energy with a capacity of 300 hp all during the year. The river flow on the average in any one year is

200 sec-ft for 2 months
100 sec-ft for 2 months
75 sec-ft for 1 month
40 sec-ft for 7 months

a. If the site is developed as a run-of-river plant without storage, what capacity must be provided in a stand-by plant? The efficiency of the hydraulic plant will be 80 per cent over-all.

b. If a reservoir could be arranged upstream, what would be the percentage of the stored water that would provide margin for evaporation and seepage? Would a stand-by plant be necessary?

25-5. The daily load on a storage hydraulic plant varies as follows:

200 kw from midnight to 8 A.M.
400 kw from 8 A.M. to 4 P.M.
800 kw from 4 P.M. to 6 P.M.
600 kw from 6 P.M. to midnight

The over-all efficiency of the plant at all loads is 80 per cent. The area of the pond is 500,000 sq ft. Starting at midnight, the headwater elevation is 1,500 ft, the tail-water elevation 1,400 ft. Assume that tail-water elevation is independent of the rate of flow. The flow entering the pond remains constant over the 24 hr at 60 sec-ft.

a. Plot the variation in headwater level throughout the day.

b. Plot the load curve on the same graph.

25-6. A 10,000-kw run-of-river hydraulic plant operates on an available head of 200 ft and has an over-all efficiency that remains constant at 90 per cent for all loads. For a given day during the low-water season the customers receiving primary power have the following loads:

8,000 kw for 2 hr
6,000 kw for 2 hr
4,000 kw for 10 hr
2,000 kw for 10 hr

For a river flow remaining constant at 525 sec-ft, how much secondary power (energy) could this plant deliver during the entire day?

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CHAPTER 26

HYDRAULIC TURBINES

26-1. Turbine Types. Water turbines convert the potential energy in water stored behind a dam into shaft work, ordinarily used to drive an electric generator. In this conversion the potential energy of the water converts to kinetic energy as it falls through the head of the plant. The water imparts most of this kinetic energy to the turbine runner to develop a torque on the shaft and produce shaft work. A shaft-speed governor regulates the opening of the turbine gates or nozzles in response to changing loads. The turbine must be firmly supported in a setting with passages to lead the water to the runner and passages to take the spent water from the runner.

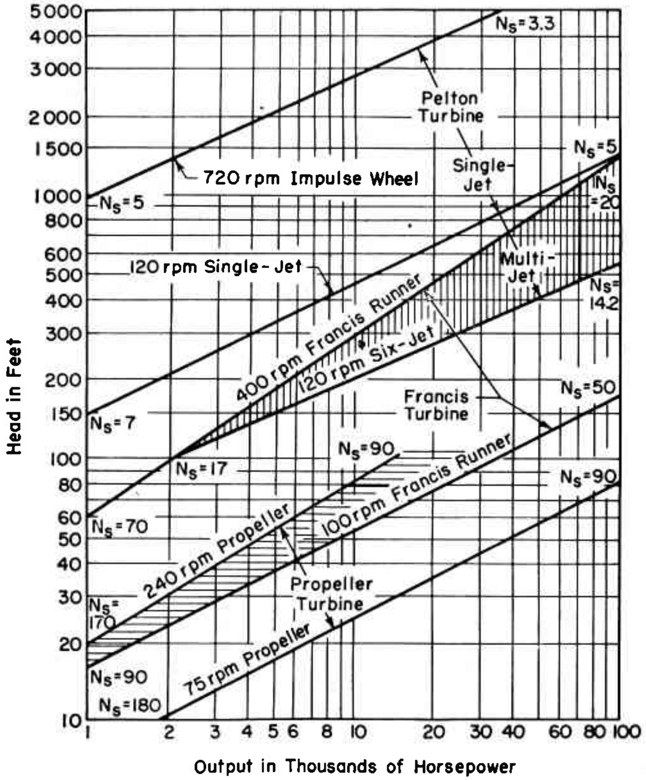

FIG. 26-1. General range of head and output for three types of hydraulic turbines classified by specific speed N_s. Shaded areas show overlapping applications of two types. (*Courtesy of V. A. Thiemann.*)

Turbines may be broadly classed as impulse and reaction types.

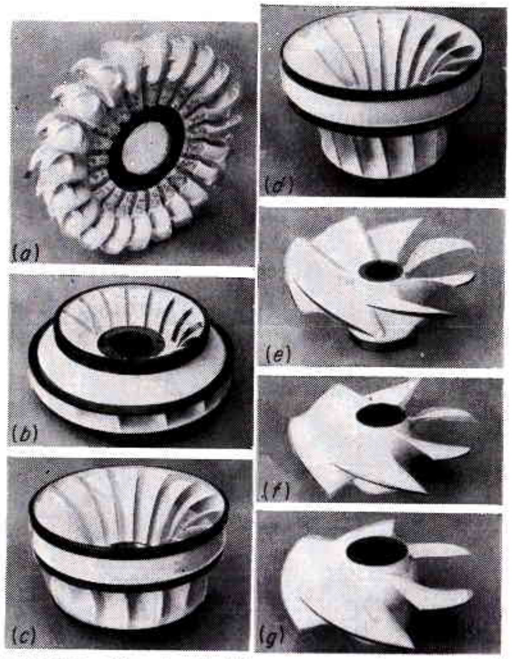

FIG. 26-2. Testing models of water-turbine runners show variation in shape with specific speed N_s as governed by water head. (*a*) Highest heads use impulse, or Pelton, wheel for $N_s = 3.5$ to 4.3. Intermediate heads use Francis runners: (*b*) $N_s = 18$ to 20; (*c*) $N_s = 54$ to 62; (*d*) $N_s = 80$ to 85. (*e*) Medium to low heads use mixed-flow runner for $N_s = 100$ to 125. (*f*) Lowest heads use propeller runner for $N_s = 110$ to 150. (*g*) Kaplan runner has pivoted blades that can be angled for best efficiency at any load for $N_s = 80$ to 150. (*Courtesy of Dominion Engineering Company.*)

Figure 26-1 shows how the various types are used for a range of heads and capacities. Figure 26-2 shows typical runner designs for the various types.

Lowest heads use *propeller*, or *axial, turbines;* the types actually used vary with the turbine capacity. The *Kaplan turbine* is a special type of propeller turbine featuring variable-angle blades that can be changed with water flow to maintain high efficiency. *Francis*, or *radial-* and *mixed-flow*, turbines are used for an intermediate range of heads. *Pelton*, or *impulse, turbines* work at the highest heads. Figure 26-1 should be regarded as a general guide, since some turbine types fall outside the limits shown.

These turbines may be further classified according to their *specific speed*, which is the rpm at which a scale model, as in Fig. 26-2, would run to develop 1 hp at a 1-ft head. The specific speed may be calculated as

$$N_s = \frac{N\sqrt{P}}{h^{5/4}} \qquad (26\text{-}1)$$

where N_s = specific speed, rpm
N = speed for maximum efficiency, rpm, at given head of actual full-scale turbine
P = power output, hp, of actual full-scale turbine
h = available head, ft, of actual full-scale turbine

Figure 26-2 shows typical runner shapes for ranges of specific speeds.

26-2. Axial-flow, or Propeller, Turbines. Figure 26-3 shows the principal elements of a low-head propeller turbine. The shaft drives a vertical electric generator in the generator room above, not shown.

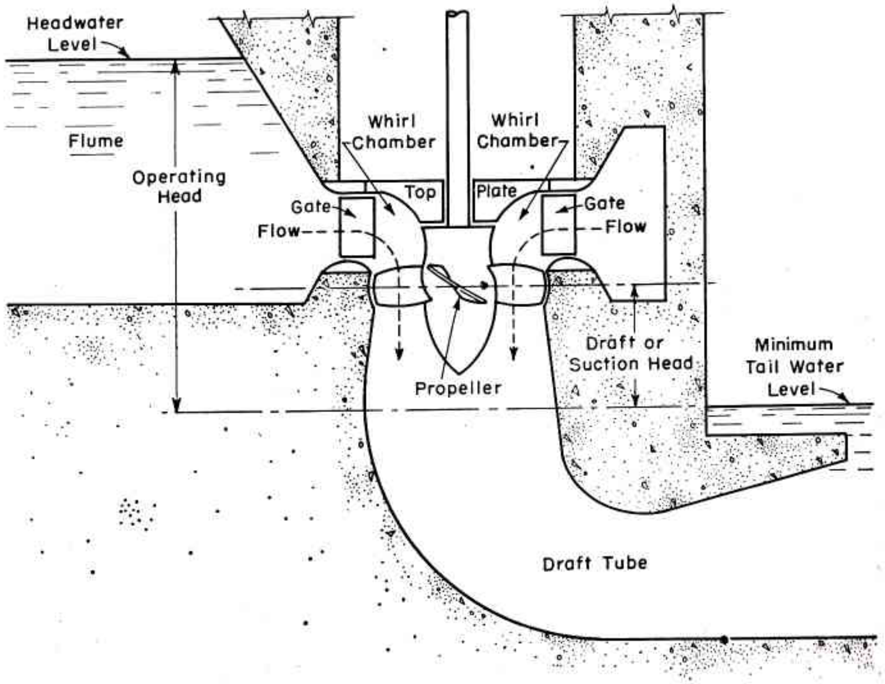

FIG. 26-3. Typical propeller-, or Kaplan-, turbine runner takes water from whirl chamber. Gates control amount of water flow and its radial-flow direction into whirl chamber. Water pressure above the runner and suction below induced by draft tube develops water force that turns impeller. (*Courtesy of V. A. Thiemann.*)

This turbine has the highest specific speed of all types, making it suitable for the lowest heads and large volume flows.

High rates of water flow develop high velocities which make cavitation a serious problem. The propeller runner may be run reversed to pump water in pumped-storage plants.

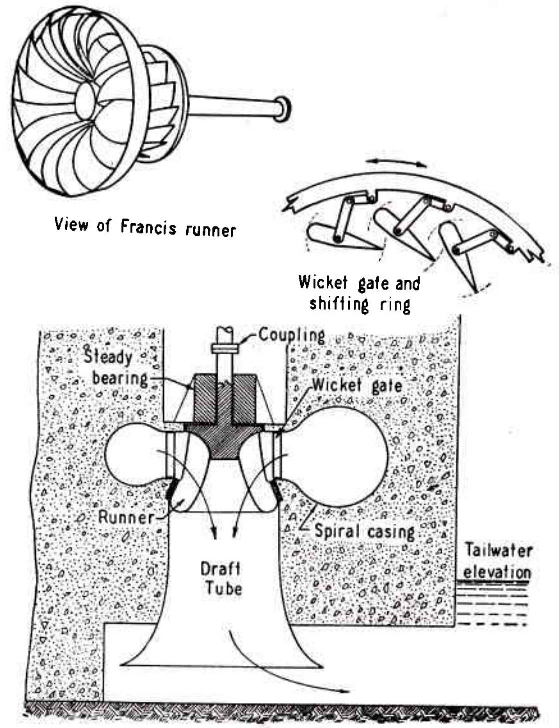

FIG. 26-4. Typical Francis turbine receives water through wicket gates from spiral casing. Water flow changes from a radial direction to a downward axial flow in runner blades.

Propeller turbines are built with: (1) fixed blades and (2) variable-pitch blades. While the fixed-blade type is cheaper to build, its efficiency drops off sharply with reduced loads (Fig. 26-7), i.e., lower water flows. They work best where they can be kept fully loaded.

The variable-pitch propeller turbines are also called Kaplan turbines. The blade pitch may be changed to get maximum efficiency for the existing load and head on the turbine. The blades may be adjusted manually by shutting down the turbine for about a half hour. For plants with more variable heads the blade pitch can be changed automatically as the controlling conditions vary.

Adjustable wicket gates in the speed ring control the rate of water flow entering the whirl chamber and runner. A thrust bearing in the generator above supports the propeller runner and shaft. The wicket gates extend all around the turbine perimeter and operate by levers attached to a gate ring, above the top plate, by links.

26-3. Francis Turbine. Figure 26-4 shows a typical arrangement of a Francis, or reaction-type, turbine. Water passages are smaller than in a propeller unit of the same capacity. The draft tube, as in Fig. 26-3 also, produces a suction head that gives some flexibility in placing the turbine above tail-water elevation without sacrificing energy. Runners usually mount vertically as shown, but some are used horizontally for high heads and small capacity. Running clearances must be kept small to reduce leakage to a minimum. Silt and cavitation erosion attack the rotor.

Wicket gates controlled by a shifting ring (Fig. 26-4) control the water flow from the spiral casing into the runner blades. A gate or valve in the headworks can shut off water feed to the spiral casing in emergencies or to unwater the turbine for repairs.

The draft tube being submerged in the tail water creates a partial vacuum at the runner outlet to maintain nearly full head across the runner. The flare of the draft tube decreases water velocity smoothly to regain part of the kinetic energy as head on the turbine.

26-4. Impulse, or Pelton, Turbines. The impulse turbine is used in high-head plants and sometimes in medium-head plants with small capacity. Figure 26-5 shows the elementary arrangement of a Pelton wheel with a single nozzle. As many as six nozzles may be used with high-capacity Pelton wheels. The specific speed per jet of a multijet unit remains constant, but the over-all specific speed of a turbine increases with the square root of the power output.

The Pelton turbine features simplicity and ruggedness with no close-running clearances and zero leakage. Parts attacked by cavitation and silt erosion can be easily replaced. Stellite and stainless-steel materials resist these destructive effects in critical areas. The Pelton turbine must be placed as close to tail-water elevation as conditions permit, and the runner must not be allowed to submerge. The draft tube cannot be used with this turbine.

A splitter divides the buckets on a Pelton wheel (Fig. 26-5) to divide the water jet into two equal parts leaving the buckets. This reduces the axial thrust on the shaft to practically zero. The same impact and momentum principles apply to the water turbine as in an impulse steam turbine.

Pelton turbines have long penstocks because of the high heads. A shutoff valve usually precedes the needle-controlled nozzle. Horizontally mounted runners may have up to two nozzles, but more nozzles usually require the runner to be mounted with a vertical shaft.

26-5. Speed and Pressure Control. Governors responding to shaft speed control the flow of water entering a turbine by automatically opening the wicket gates or nozzle needles. On a Kaplan turbine the governor also positions the blades for existing flows and heads to maintain optimum efficiency. On a Pelton turbine the governor also operates auxiliary relief valves or jet deflectors in addition to the nozzle-needle position.

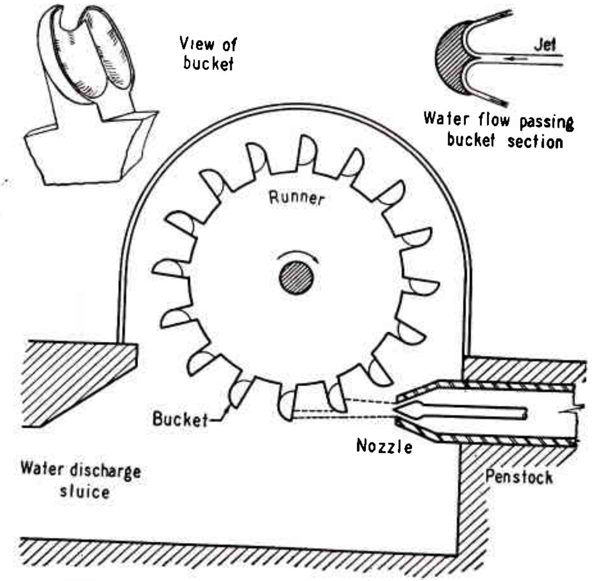

FIG. 26-5. Runner of Pelton (impulse) turbine receives high-velocity water jet from nozzle to react with buckets and develop high torque on shaft. Axial movement of needle in nozzle orifice controls rate of water flow.

Long penstocks on high-head Pelton and Francis turbines carry heavy masses of flowing water. On sudden load changes these cannot be stopped or accelerated quickly without setting up destructive water hammer that can burst the penstock. Surge tanks connected to the lower end of the penstocks, above the powerhouse, receive water from the penstock, to decelerate it slowly during sudden load drops, or feed water into the penstock smoothly on sudden load increases.

Pressure regulators below the surge-tank connection for Francis turbines limit the pressure rise on sudden changes. These relieve water from the turbine casing to discharge it to the tailrace. In water-wasting arrangements the regulator opens as fast as the wicket gates close, or vice versa. This holds penstock pressure constant and gives a constant water flow through the powerhouse. In water-saving arrangements the regulator opens on sudden load drop and then gradually closes as the governor operates the wicket gates. Some pressure regulators discharge into an energy absorber that dissipates the jet energy in turbulence.

In Pelton turbines auxiliary relief nozzles divert water from the power nozzle as the governor quickly decreases flow through it. In multi-nozzle turbines a deflector plate, or collar, swings into the water jet from each nozzle and deflects some water from the runner buckets. This reduces turbine output to the load immediately. The needles then smoothly reduce water flow while the deflectors move out of the jet streams to keep the turbine output constant at the new lower level.

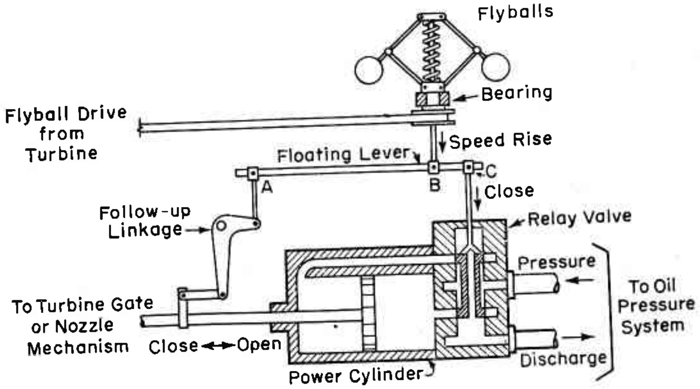

FIG. 26-6. Simple governor has flyballs responding to speed, relay valve, or mechanical amplifier, cylinder and piston to operate gate mechanism, and a follow-up system to close relay valve at proper time and prevent hunting. (*Courtesy of V. A. Thiemann.*)

Figure 26-6 shows a simplified arrangement of a water-turbine governing system. The flyball position depends on the speed of the turbine shaft. The flyballs may be belt-driven, as shown, or driven by a small motor energized by a small generator coupled to the main turbine shaft. A spring-tension adjustment on the flyball shaft permits changing speed or turbine output. A relay valve and power cylinder amplify the small force created by the flyballs to control the forces needed to operate the turbine gates.

The follow-up linkage resets the relay, or pilot, valve after the power piston has adjusted the position of the gates. Some governors have two-stage amplification to increase sensitivity and force output.

On small governors the power cylinder is part of the governor; on large governors the power cylinder, called a "servomotor," mounts on the turbine while the governing and control mechanisms, called an "actuator," mount in a cabinet on the operating floor. Oil pressure operates the power cylinder and is produced by an oil pump and contained in oil-pressure and -discharge tanks. Governors may have limit devices to prevent the gate opening from exceeding a set amount.

On units operating alone the governor usually holds constant speed at all loads. For two or more units the governors may be set to drop speed as much as 2 to 5 per cent when load changes from zero to full. This helps to divide the load in a prearranged schedule among units, automatically and without hunting. The governor spring tensions are varied to maintain rated speed; they also control the division of total load among the units in the station.

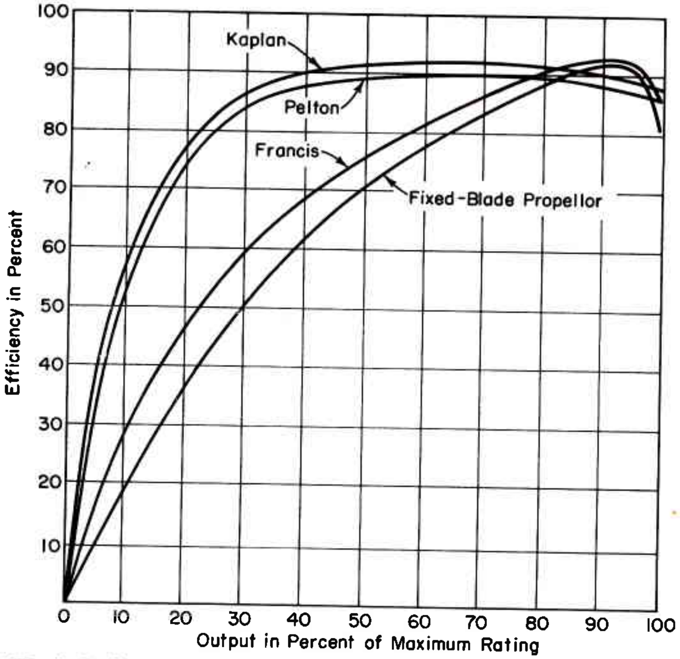

FIG. 26-7. Typical efficiencies for water turbines when running at constant speed and head. Kaplan and Pelton turbines perform well at part loads compared with Francis and propeller turbines.

Hydraulic turbines are designed to be able to run at runaway speed (no load and full water flow), which is about twice normal speed. This eliminates the need of an overspeed trip in the governing system.

26-6. Turbine Performance. Turbine performance is referred to the head of water available at the turbine gate or inlet valve. This excludes losses in the intake system but charges the turbine with losses in the scroll case and draft tube. Equation (25-1) can also be used for turbine power output when the efficiency is the turbine efficiency and h is the head at the turbine inlet.

Figure 26-7 shows some typical efficiency curves for the principal types of turbines. Turbine operation at different conditions or for different-sized but geometrically similar units may be estimated by these similarity equations:

$$\frac{Q_1}{N_1D_1^3} = \frac{Q_2}{N_2D_2^3} = \text{constant} \tag{26-2}$$

$$\frac{h_1}{N_1D_1^2} = \frac{h_2}{N_2D_2^2} = \text{constant} \tag{26-3}$$

These equations together with the specific-speed equation may be used to develop other relations between the variables.

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CHAPTER 27

HYDRO POWER STATIONS

27-1. Hydro Projects. Most hydro projects are developed to produce electric power by privately owned utilities. Many government-owned projects attempt to serve several purposes in addition to the generation of power, such as: (1) river flood control, (2) maintenance of water levels downstream for navigation needs, (3) storage of irrigation water, and (4) storage of drinking-water supply.

The layout of hydro-station projects depends on the services the station must perform, the surrounding topography, the controlling economics, and the likelihood of developing secondary-power capacity. These factors in turn, together with the physical characteristics of the site, govern the type and arrangement of dams, spillways, and conduit systems. Many types of dams may be found in one project, the economics of the particular location largely dictating the type chosen. High dams predominate where the river falls over a long distance, and low dams most often are erected at the head of concentrated falls.

Dams close to the head of waterfalls cost less but often have longer conduits to the powerhouse. The powerhouse usually stands at the foot of the rapids to minimize excavation costs. Long conduits cause pressure losses and raise the installation cost.

Most hydro stations have buildings enclosing the generators and the powerhouse crane that handles the main unit parts (Fig. 27-3). The crane must be high enough to clear the generators and be able to pull up the turbine runner out of the spiral casing and place it on the operating floor for maintenance. The building also houses the control room, shops, offices and ancillary rooms. Switchgear and transformers are usually outdoors.

Some stations have been built with outdoor operating floors. With no building for protection all equipment must be clad with a weatherproof casing, offsetting somewhat the saving in building investment. A gantry crane straddles the generators and runs on rails on the outer edges of the operating floor.

Some stations compromise with a semioutdoor arrangement. The generators are covered by low buildings just clearing the tops of the generators. A gantry crane straddles the buildings on the outside and gains access to the equipment by removing hatch covers in the roofs above them.

A few stations are completely underground. In these the topography makes it more economical to hew the generator room, the penstocks and

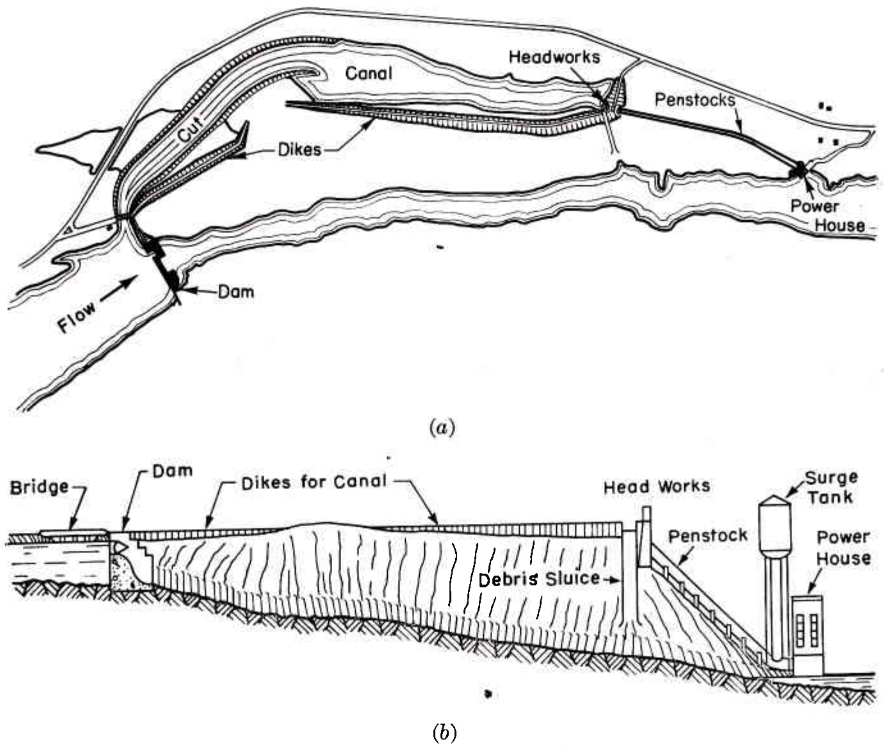

FIG. 27-1. (*a*) Plan view of medium-head plant to illustrate hydro power system. Water is brought to the powerhouse by canals and a penstock. Surplus water is discharged by the spillway into the original river channel. (*b*) Profile of system. (*Courtesy of V. A. Thiemann.*)

tunnels, and the tailrace out of the rock. The surge tank may be a vertical shaft driven to atmosphere through the rock.

Most hydro stations run directly under manual supervision by operators in a control room. Much of the equipment runs automatically, even to the extent of starting units or taking them off the line and shutting them down. Several small stations on a common river may be fully automatic in operation without manual attendance. They would be telemetered to a central control board at one of the stations manually supervised. They usually have alarm systems that warn operators of trouble and automatically shut down equipment in emergencies. These

stations might be inspected daily or weekly for signs of operating trouble. Some stations might be remotely controlled from a central control board and be semiautomatic in operation, having automatic shutdowns for emergencies but not automatic starting cycles.

27-2. Hydro Plants. Because of the many factors governing the design of a hydro plant, each one is unique, all factors never being the same for any two plants. Figure 27-1 shows the plan and elevation for a small medium-head hydro plant. The main dam across the river forms a modest pond upstream and diverts the water into the canal paralleling the rapids in the river downstream of the dam. The mile-long canal runs in an excavated cut from one side of the dam into a natural depression to the headworks dam above the plant. Dikes on the riverside of the canal form a built-up bank. At the headworks dam two main gates control the water flow entering the two $\frac{1}{4}$-mile wood-stave penstocks taking water down to two turbines in the plant building. The surge tanks connect to the foot of the penstocks and project above the level of the canal. A debris sluice at the river end of the headworks dam gives a bypass for discharging floating matter to the rapids below and keeping it out of the penstocks and turbines.

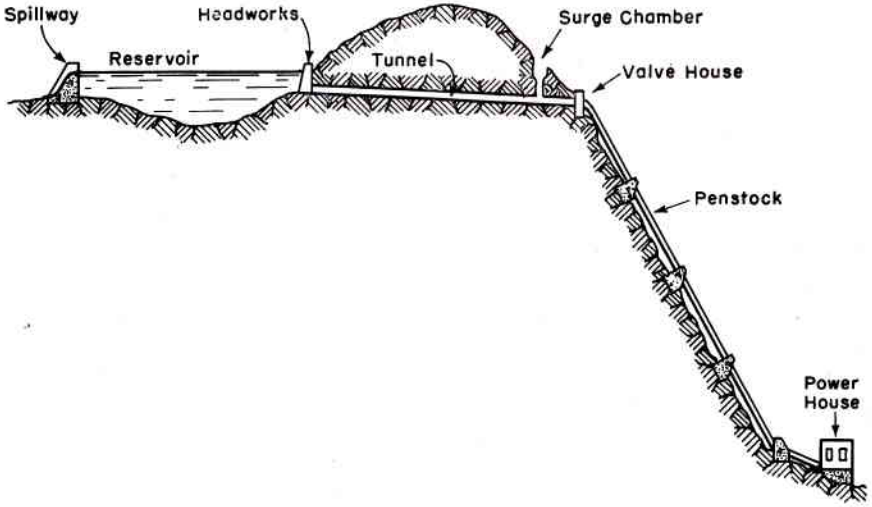

FIG. 27-2. High-head power plant. Note the separation of main dam, spillway dam, and powerhouse. This type of site might also be used for an underground station.

The spillway at the main dam controls the level of the water in the head pond upstream and the level in the canal leading to the headworks.

Figure 27-2 shows the elevation of a high-head plant; the dam at the left has a spillway that might discharge to another river than the one at the powerhouse tailrace. The surge chamber stands before the valve house and after the tunnel from the headworks. Butterfly valves in the valve house at the head of the penstocks and gate valves at the turbines in the powerhouse control the water flow. This type of arrangement might be suitable for an underground power plant where the rock formation might be weak at the river edge below, or there might be danger of rock slides, or tunneling might be cheaper than penstock construction. The powerhouse would then be at the same elevation, and an underground tailrace would discharge to the river at the right, not shown.

Figure 27-3 shows a section through a low- to medium-head plant using Kaplan turbines. The powerhouse forms part of the dam, making the headworks, inlet conduit, turbine and scroll case, and tailrace an integral unit. Gantry cranes over the tailrace and intake control the gates for unwatering the turbine for inspection and repair. Control room and offices are on the top floor of the powerhouse building, and the electrical equipment stands on the roof.

Figure 27-4 shows the plan view of the famous Hoover Dam on the Colorado River between Nevada and Arizona. This backs up the river to form Lake Mead, some 150 miles long, extending to the outlet of the Grand Canyon in Arizona. The arch dam creates a head of under 400 ft and stands upstream and adjacent to the two power-plant wings housing 17 turbine generators. Four intake towers, each having two cylinder gates, control flow into the four main penstocks laid in tunnels. Two spillways, controlled by four drum gates each, pass flood flows round the dam through spillway tunnels to points several hundred feet downstream of the plant. The discharge of Hoover Dam Plant forms the headwater of the Parker Dam reservoir, the next plant some hundred miles downstream.

27-3. Plant Auxiliaries. Many hydro plants have small auxiliary turbine-driven generators to supply auxiliary power for the plant when it is shut down. Some plants may have internal-combustion (i-c) engine-driven exciters for emergency stand-by so that large units can be started when the entire plant is shut down.

Governor oil systems must have enough power to open gates to start turbines from standstill. If the main oil pumps are gear-driven off the main turbine shaft, there must be separate motor- or engine-driven pumps to raise oil pressure during shutdown.

Hydro stations may have a wide variety of auxiliaries: circulating-water pumps for bearing cooling water, generator cooling, and transformer cooling; and air compressors for generator brakes, for agitating intake water to prevent ice formation on headworks valves and conduits, and for air pressure to depress tail water below the turbine when the generator acts as a synchronous condenser.

Air brakes on generator shafts are set when the turbine is at standstill,

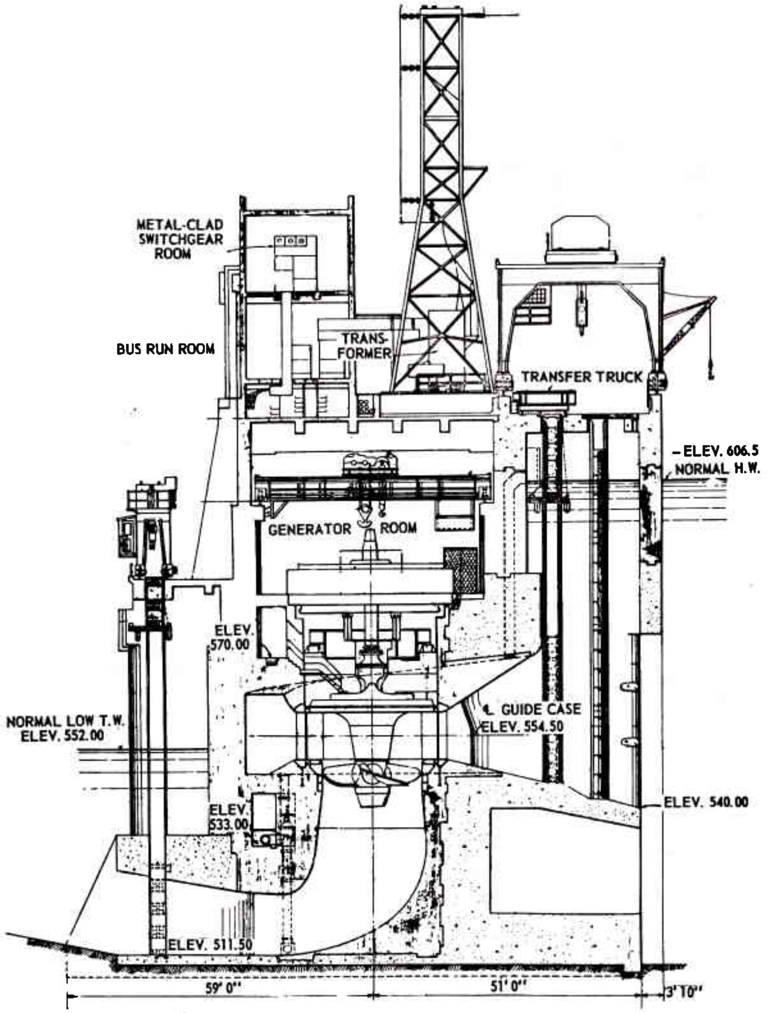

FIG. 27-3. Typical low-head Kaplan type of turbine at the Rock Island Plant on the Columbia River of Public Utility District No. 1 of Chelan Co., Washington. Six units were recently installed rated at 34,000 hp each, 100 rpm, 45-ft head. Plant now has 10 units total. *(Courtesy of Allis-Chalmers Manufacturing Company.)*

and not to slow the generator down when shutting down. Electric power must be supplied to: pumps for water supply and drainage, and for unwatering turbine casings prior to repair and inspection; motors driving transformer fans and oil pumps; motors on the station crane and hoists for spillway and head gates, and penstock valves. The station needs electric power for lighting, heating, and ventilating and for storage-

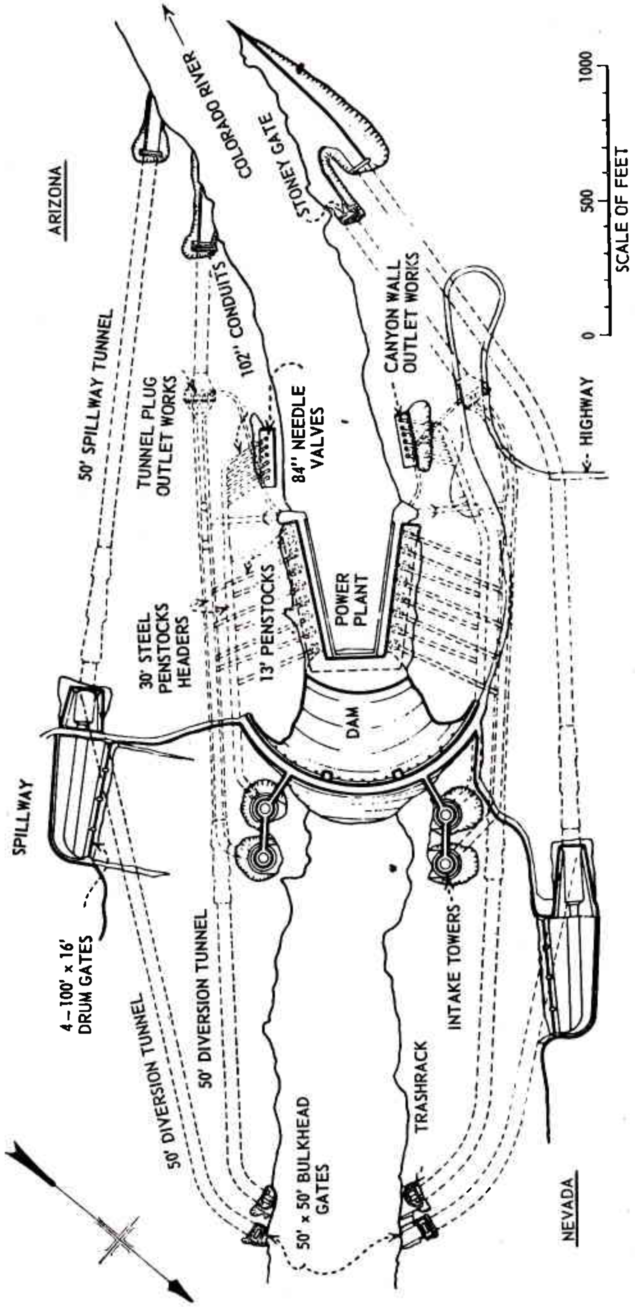

FIG. 27-4. Over-all plan view of Hoover Dam showing the essential details, including the twin spillways, intakes, tunnels, and penstocks. The installation consists of 15 Francis-type units of 115,000 hp each and one unit of 70,000 hp at a design head of 480 ft. A seventeenth unit to drive a 100-mva generator will start running in 1961. (*Courtesy of Allis-Chalmers Manufacturing Company.*)

battery charging. The battery supplies power for switchgear control and may furnish emergency light and power.

27-4. Plant Operation. The operation of run-of-river and storage plants was described briefly in Sec. 25-5. Run-of-river plants can only generate energy as river flow becomes adequate. Storage plants vary in degree; those with small pondage may be able to store the river flow during low loads in a week to help carry the peak loads in that week.

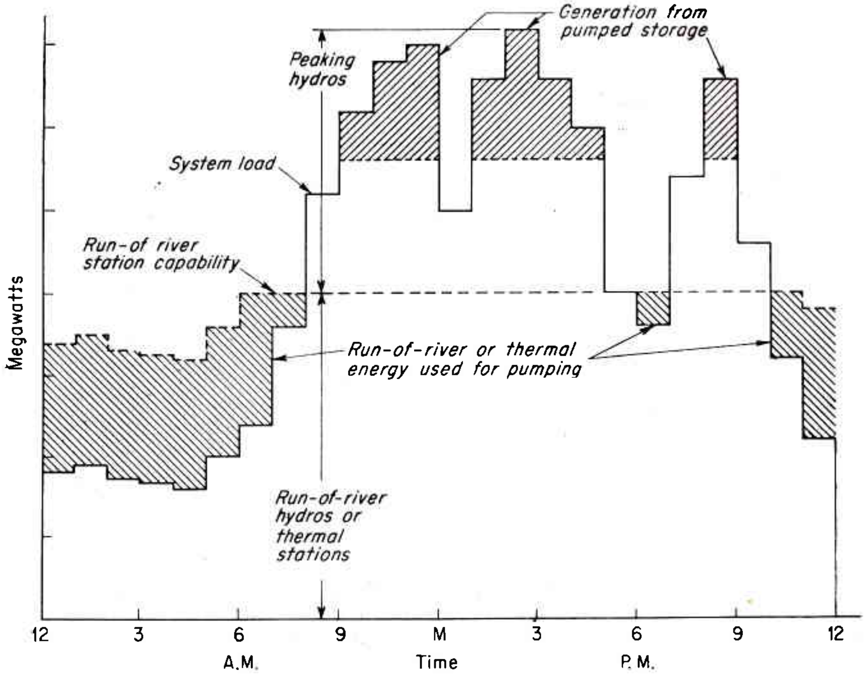

FIG. 27-5. A daily load curve illustrating use of pumped storage for load peaks. System load is shown by solid lines. Power for pumping may come from thermal or other hydro stations.

Storage plants with large upstream reservoirs may have enough water during dry summers to carry normal loads. Again, situations vary widely so that each plant is unique.

In some yearly cycles most water plants will be spilling water during high river flows and low-load periods. Some of this excess energy can be stored in *pumped-storage plants*. The simplest concept would be high-level and low-level reservoirs connected through a penstock with a generating-pumping plant at the lower end.

During light system loads the excess generated energy would be used to energize a motor-driven pump and pump water from the lower to the upper reservoir, in that way storing energy. When the system load exceeds the system capacity, water from the upper reservoir would be discharged through a turbine generator in the pumped-storage plant to release energy and help carry the system peak loads. During the next light-load period the excess energy would again be used to return water from low-level to high-level reservoir.

Figure 27-5 shows how this operation works in conjunction with the daily load variation in a system. Such a pumped-storage plant can be used with steam-electric and i-c engine plants also. This helps to keep the thermal plants more uniformly loaded for better efficiency. If the reservoirs are large enough, this scheme can be operated on a seasonal basis as well as a daily or weekly basis.

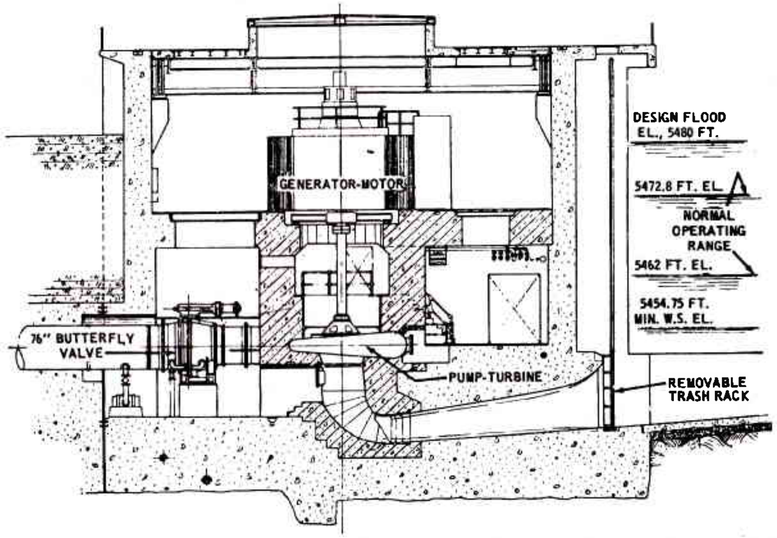

FIG. 27-6. Reversible pump-turbine as used in the Flatiron Plant. As a turbine it develops 12,000 hp at 290-ft head at 257 rpm; as a pump it delivers 370 cfs at 240-ft head at 300 rpm. Note the butterfly valve to control the flow of water.

This system of energy storage does not work at 100 per cent efficiency: some energy is lost in the generating and pumping equipment and in electric transmission. Also, some of the water pumped evaporates in the upper reservoir, diminishing the stored energy. The over-all efficiency of energy supplied to load compared with energy initially generated runs about 60 per cent. Despite this high loss this scheme is finding increasing application because of capital savings that can be made in some situations.

Older pumped-storage plants had separate motor-driven pumps and turbine-driven generators. Later plants had the pump and turbine on a common shaft with the electrical element acting as either generator or motor. In the latest plants this has been simplified to two elements: (1) the generator-motor electrical element and (2) the pump-turbine mechanical element (Fig. 27-6). A Francis type of runner acts as a normal turbine turning in one direction and as a pump when turning in the other direction.

27-5. System Plant Operation. Most modern power systems use both hydro and steam stations to supply the total load. At times of low water flows the hydro plants function most effectively as peak-load units and vary their output to meet the fluctuations in load. This keeps the thermal plants at constant load, which produces a better efficiency.

During periods of high water flows the thermal plants and hydro stations reverse their roles, the hydro taking the base load and the thermal plants the peak load. But even in this operation the hydro plants would vary their outputs to meet the load fluctuations. Hydro turbines can be started from "cold" in a few minutes, while steam units need about 6 to 8 hr.

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CHAPTER 28

ENERGY-SOURCE DEVELOPMENTS

28-1. Energy Supply. Our modern civilization, often called the machine age, is more truly the energy age. Through the ages man has successively devised more and more complex machines to make things and perform services. Man could succeed with this program only as he discovered sources of energy to power his machines. The growing energy demands, as discussed in Chap. 1, suggest that one day we shall exhaust our known energy resources. This might be disastrous, for the living standards of the nations of the world vary directly with their annual energy consumptions.

Today the peoples in North America and Europe use most of the world's total energy. They are less than 30 per cent of the world's population. Other nations are now embarking on vigorous industrial-development programs that will accelerate growing annual energy demands. This potential exhaustion has sparked many organizations and governments to search for other sources of energy and devise new and more efficient ways of using energy.

28-2. Energy Sources. As Fig. 1-1 shows, wood was the major source of energy in the United States up to about the Civil War, when coal came into wider use. In those early days wood supplied our thermal needs, and we had hydro power and wind power, as well as draft animals, to supply much of our mechanical work. Steam power was in its infancy and did not consume appreciable amounts of energy in the form of wood fuel. Wind, water, and wood can be regarded as continuous or replenishable supplies of energy. In terms of mankind's tenure on this earth, these energy sources will always be available.

But since the Civil War we have found the fossil fuels coal, petroleum, and natural gas to be much more plentiful and convenient to use. Today oil supplies the major part of our energy, with coal and gas evenly supplying the remainder and with only token amounts coming from hydro, water, wood, and solar energy.

Besides practical convenience the use of a fuel will be governed largely by the cost of winning it and preparing it for use. As fuels grow scarcer, we undoubtedly can afford to pay more for those which are buried deeper and need more preparation and transportation to market.

The brightest hope for the immediate future lies in our deposits of uranium and thorium. The ultimate development of fission reactors to use these nuclear fuels will multiply our fuel resources by about forty times on a world-wide basis. They about triple the known energy resources of the United States.

The long-range promise lies in using deuterium, heavy hydrogen, as the fuel in fusion reactions. Deuterium can be separated from natural waters, and the amount available is so vast that the energy potential can be regarded as limitless.

28-3. Solar Energy. The heat energy we receive from the sun has long been used in tropical regions for heating water and drying certain types of crops. Scientists have always been interested in the sun and its energy yield on earth. Experiments are now under way to use this energy for power production, house heating, air conditioning, cooking, and high-temperature melting of metals. The large-quantity use of solar energy poses the problem of needing large heat-absorbing areas to capture it. In addition solar energy is available only for a part of a day, and cloudy and hazy atmospheric conditions cut the energy yield sharply.

The upper atmosphere of the earth receives solar energy at a rate of 442.4 Btu per sq ft-hr ± 2 per cent. The energy at the earth's surface is

$$Q_a = S\eta_a \qquad (28\text{-}1)$$

where Q_a = energy received at earth's surface, Btu per sq ft-hr
S = solar constant, 442.4 Btu per sq ft-hr
η_a = efficiency of energy transmission through atmosphere, fraction

In the United States Q_a averages 250 to 350 Btu per sq ft-hr. The transmission efficiency depends on four factors: (1) air mass, (2) the sun's altitude, (3) clouds and haze, and (4) diffuse radiation. The air-mass factor depends on the latitude, time of year, and time of day. The altitude factor depends on the angle between the sun's rays and the receiving surface. The cloud and haze factor depends on meteorological conditions and pollutants in the air. The diffuse-radiation factor accounts for the nondirect radiation reflected from the sky to the earth.

Solar energy consists of electromagnetic radiations. Wavelength varies from 0.3 μ to about 20 μ. About 99 per cent of the energy lies in the range from 0.3 to 3 μ, with a peak at about 0.5 μ.

Solar radiation may be converted to two types of useful energy forms: (1) heat and (2) light. In general

$$Q_o = Q_a\eta_o \qquad (28\text{-}2)$$

where Q_e = useful part of incident energy, Btu
Q_a = incident energy, Btu
η_e = efficiency of collection equipment, fraction

Efficiency usually falls with rise in collection temperature. Two types of collectors are used to convert solar energy to heat: (1) flat plate and (2) concentrator. Figure 28-1 shows these in diagrammatic form. Newer flat-plate collectors have the coolant tubes integral with the collecting surface. Covering the collecting surface with multiple sheets of glass and evacuating the intervening space raises efficiency and reduces heat loss. Flat collectors can generate steam at low temperatures.

Concentrating collectors may be used to generate medium-pressure steam. They use many different arrangements of mirrors and lenses to concentrate the sun's rays on the boiler. This type shows better efficiency than the flat-plate type. For best efficiency collectors should be mounted to face the sun as it moves through the sky. The arrangement shown in Fig. 28-1 shows how collectors may be used to heat fluids or generate steam.

Solar energy in the light range 0.4 to 0.8 μ of the spectrum may be used in photochemical and photoelectric processes. The photochemical effect produces photosynthesis in vegetation, the process that converts solar energy to chlorophyll and organic matter. The process takes the form

$$XCO_2 + XH_2O + \text{solar energy} = X(\text{CHOH}) + XO_2 \tag{28-3}$$

where (CHOH) represents a carbohydrate.

We can use vegetation to produce thermal energy by combustion, sometimes by forming intermediate fuels such as alcohol and gasoline. Combustion reactions proceed in the reverse direction of Eq. (28-3).

Photovoltaic cells convert solar rays to electrical currents. Selenium, as an example, when struck by light rays has its electrons excited to the degree that they leave the metal and travel as a minute electrical current through a connected circuit. Modern photoelectric cells using silicon wafers, called "solar cells," convert solar energy to electrical currents with an efficiency of 10 to 12 per cent. These cells produce potentials and currents suitable for charging storage batteries.

28-4. Wind Power. Energy in the wind has been used for many centuries to sail vessels, pump water, and grind wheat and corn. Winds essentially are created by solar heating of atmospheric air.

Several attempts have been made since 1940, and current development goes on, to use wind to generate electrical energy. Wind energy stems from two prime effects: (1) heating of the atmosphere that generates convection currents and (2) relative motion of the atmosphere to the

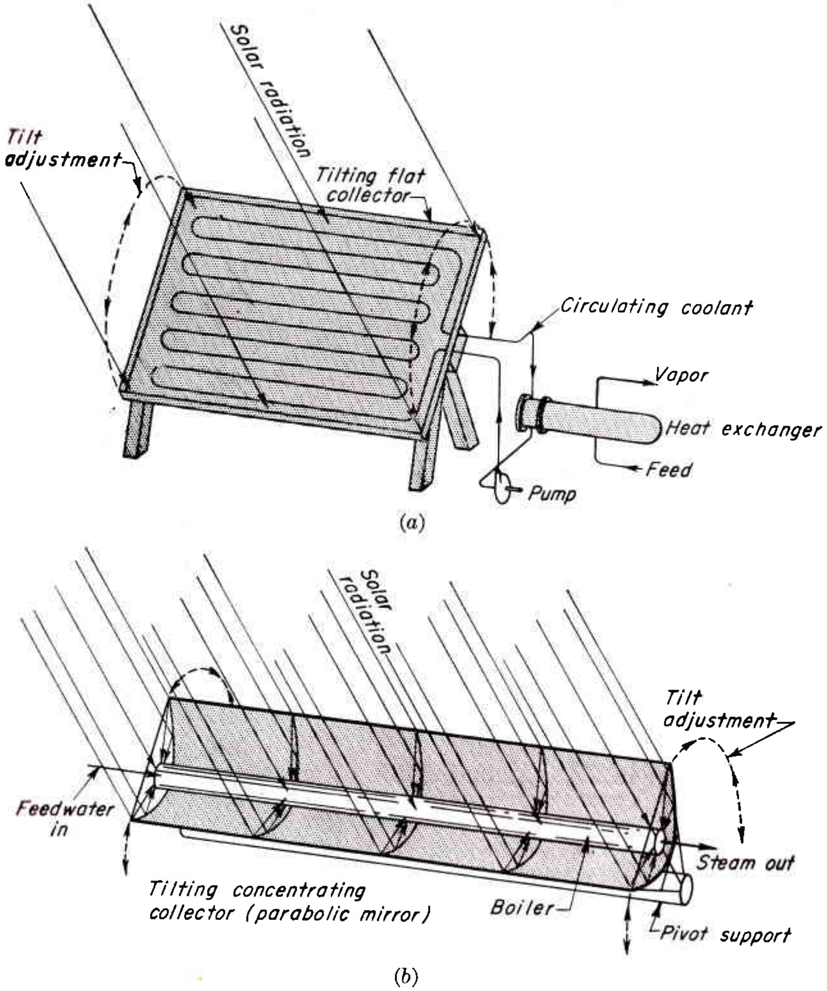


FIG. 28-1. (a) Flat-plate solar-energy collector heats up coolant circulating through tubes; (b) concentrating collector focuses sun's rays on boiler at focal point.

rotating earth. Winds follow a global pattern but vary markedly in day-to-day intensity and duration at any point on the earth.

Figure 28-2 shows a duration curve for wind speeds during a year at one geographical location. Storm and gale winds prevail for only a small percentage of the time. The same is true of dead calms, when no air moves at all. The corresponding power available in the wind is shown by the concave-upward curve. It relates to wind velocity as

$$P_w = \frac{\rho A V^3}{2g_c \times 550} \tag{28-4}$$

where P_w = power in wind, hp
ρ = air density, lb_m per cu ft
A = cross-section area of stream, sq ft
V = wind velocity, ft per sec
g_c = constant = 32.2 lb_m-ft per lb_f-sec^2

Winds less than 20 mph have little value for power generation, but above 30 mph their force makes it difficult to tap with reasonable structures.

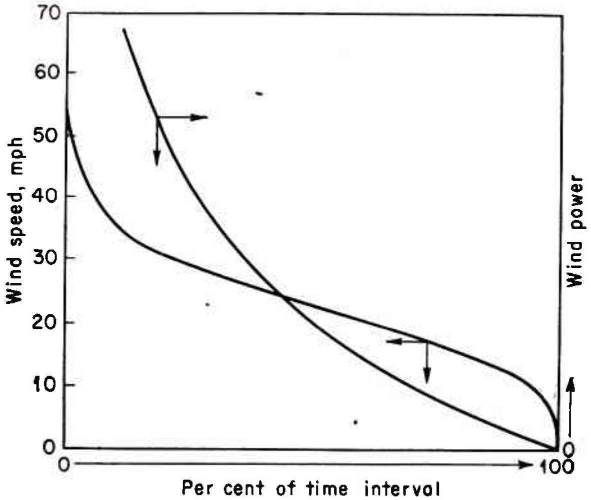


FIG. 28-2. Annual wind and wind-power duration curve for a given site. Wind power varies with the cube of wind velocity.

Power converted from wind is

$$P_t = P_w \eta_w \tag{28-5}$$

where P_t = power output from wind turbine or mill, hp
η_w = efficiency of conversion, fraction

The efficiency of conversion varies with wind velocity. A propeller type of rotor has a maximum theoretical value of about 0.60 but actually develops about 0.40.

The low density of winds means that large volumes must be handled to generate significant amounts of power. A wind turbine, or even a battery of them, has no basic effect on the surrounding territory. Wind-duration curves (Fig. 28-2) show that winds can deliver fairly constant amounts of power annually. But the wind variability means that the power delivered will also be erratic. Average wind speeds for different localities vary from 2 to 50 mph and may vary ±10 per cent from year to year.

Wind turbines would be very costly if designed to run at all wind speeds. They usually start running at wind speeds just enough to overcome the system losses and develop full power at the prevailing speed for the locality. Above this speed the wind turbine spills some of the air to stay at its rating. At some maximum speed the wind turbine is shut down and the blades feathered to let the wind pass without exerting turning forces.

Figure 28-3 shows a variety of wind-driven rotors for generating shaft power. Capacities of these may range from several hundred watts to 1,600 kw, the maximum built to date. Designs for units higher than this have been proposed.

The rotors of Fig. 28-3*a* and *b* may have a different number of blades from that shown. The blades change the direction of the wind impinging on them, developing a reactive force that moves the blade and slows up the air, which gives up part of its kinetic energy.

The unit in Fig. 28-3*c* has hollow propeller blades, which are moved by the wind. The centrifugal force ejects air from inside the hollow blades through orifices at the blade tips. This induces an air flow into inlet ports near the bottom of the hollow mast. The upflowing air passes through the nozzles and buckets of an air turbine, developing the shaft power of the unit. The unit built for the British Electricity Authority develops 100 kw with a 30-mph wind turning the propeller at 100 rpm. The low-speed air turbine drives an induction alternator through step-up gears.

The panemone rotor of Fig. 28-3*d* may be designed in many forms, such as the anemometer cup for measuring wind speeds. The Savonius rotor shown has a maximum conversion efficiency of about 33 per cent. It has the capacity for driving small water pumps.

The squirrel-cage rotor windmill of Fig. 28-3*e* was designed by the Bell Telephone Laboratories. The outer stationary blades direct the wind, whatever its direction, into the rotating blades for best efficiency. This design is expected to be good for capacities up to 1 kw and more. The rotor may be supported by the shaft from above or below. The units will be suitable for battery charging.

Table 28-1 lists some proposed and actual wind-power plants with methods of construction and operating control. During World War II a 1,250-kw wind turbine with a two-bladed rotor ran for a few months on top of Grandpa's Knob in Rutland, Vt. Fatigue failure in one of the members supporting one of the blades wrecked the unit. Repair materials were difficult to get during the war. After evaluating the cost

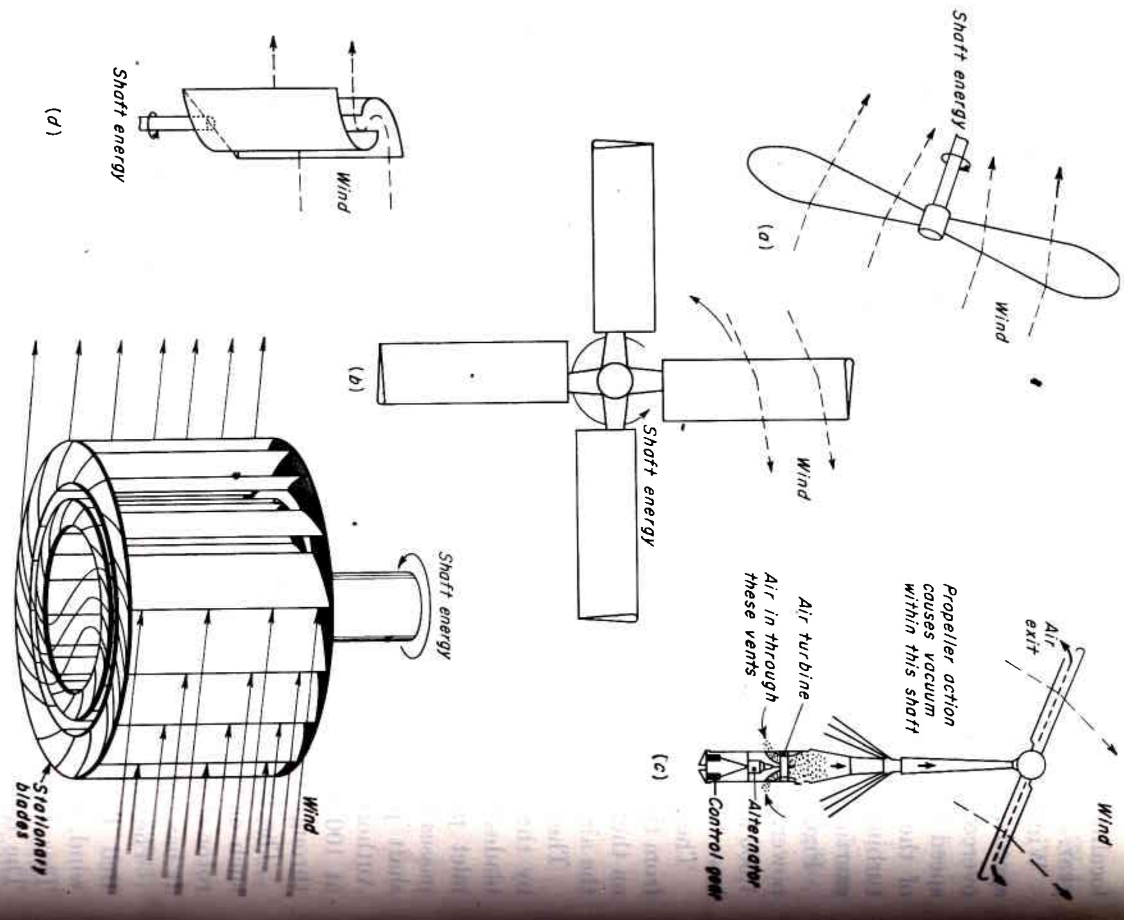


FIG. 28-3. Wind turbines or -mills vary in type of rotor used. (*a*) Two-bladed propeller; (*b*) four-bladed rotor with variable pitch; (*c*) two-bladed hollow rotor induces air flow through turbine in mast; (*d*) panemone or Savonius rotor; (*e*) squirrel-cage rotor.

TABLE 28-1. TYPICAL WIND-TURBINE PLANTS[a]

Builder or designer	Rotor diameter, ft	Rated wind speed, mph	Generator type and rating, kw	Rotor design	Control of speed or output	Optimum tip-speed ratio	Rotor speed, rpm	Hub to ground, ft	Method of weather cocking
Russia, Balaklava	98	24.6	Induction 100	3 blades	Variable pitch by flaps	4.75	30	76	Tail vane and electric drive
Proposed United States, P. H. Thomas	200	34	D-c to a-c 7,500	Twin, 3 blades each	Speed control through generator field	9	Variable, 42.75 max	475	Electric drive
Proposed, U.S. War Production Board	200	30	Induction 1,500	2 blades	Blade pitch by electro-mechanical system	12	50	150	
Smith-Putnam	175	30	Synchronous 1,250	2 blades, rectangular	Blade pitch by hydraulic system, flyball governor	6	29	110	Yaw-vane hydraulic servo-mechanism and yaw motor
Holland, Ministry of Fuel and Power	225	35	Induction 3,760	2 blades, 2-stage taper, 11½° twist, fixed coning	Blade pitch by aileron. Start-stop by pilot windmill	9.74	42.5	135	Fantail coupled to bogey wheels through centrifugal clutch on fluid flywheel
John Brown, England	50	35	Induction 100	3 blades, tapered, untwisted, free to cone and drag	Blade pitch by hydraulic control	6.5	130	78	Automatic electric control
Enfield Cables	80	30	Synchronous 100	2 hollow blades	Blade pitch by hydraulic system, variable coning	.....	Variable, 95.4 max	100	Self-oriented, assisted by wind-sensitive power-control system
Denmark, SEAS	43	28	Induction 45	3 blades	Blade-tip rotation, spring-controlled	5.4	56	66	Yawing-vane and electric-motor drive

[a] Adapted from E. W. Golding, "Electrical Energy from the Wind," Electrical Research Association, England.

and potential of the wind turbine in the United States for power generation, the builders decided to abandon the project.

Wind power proves practical for small power generation in isolated sites. But for maximum flexibility it should be teamed with other methods of power production to ensure continuity of service.

28-5. Tidal Power. At certain locations on the earth with favorable underwater topography tides may change as much as 60 ft in level within 12 hr. These significant level differences of vast masses of water suggest trapping the water to make it flow through a fall and generate shaft power.

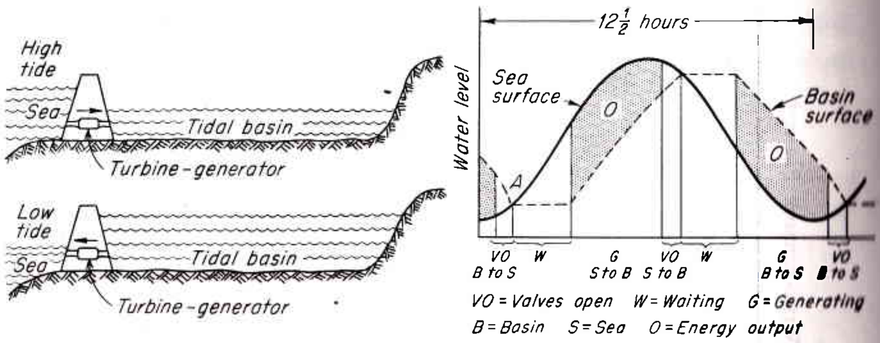


FIG. 28-4. Single-basin tidal power plant uses reversible water turbine to develop power on inflow and outflow. Graph shows operating schedule for power generation on tide rise and fall.

The Passamaquoddy project to generate power from 50-ft tides in the bay between Maine and Canada was seriously considered in the 1930s. But the remoteness of large power-consuming areas and the cost of construction and transmission made the project impractical then. Studies have recently been reinitiated to determine its economy under present conditions.

The LaRance project in France will be completed by 1963 and will have an ultimate rating of 342,000 kw. This will use a single basin and submerged reversible propeller-type turbine generators that can generate power with the water flowing in either direction through the turbine runner.

Figure 28-4 shows the schematic layout of a tidal power plant using a single basin. The graph shows the sequence of operation of the plant through a little more than one tide cycle.

When the incoming-tide sea level and tidal-basin level are equal, the turbine conduit is closed. The sea surface rises, and about halfway to high tide the turbine valves are opened and the sea flows into the basin through the turbine runner generating power. This also raises the level of the basin. The turbine continues to generate power until the tide passes through its high point and begins to drop. The water head then quickly diminishes till it is not enough to supply the no-load losses.

Bypass valves then quickly open to let water into the basin to gain a maximum water level. When sea and basin water level are again equal, the valves are closed as well as the turbine conduit. The basin level then stays constant while the tide continues to go out. After sufficient head has developed, the turbine valves are again opened and the water now flows from the basin to the sea to generate power, dropping the basin level. The plant continues generating until the tide reaches its lowest level and begins rising. Again a minimum head will be reached when it pays to shut down the turbine and open the bypass valves to drain the basin to the sea. As soon as sea and basin levels are equal, the plant repeats the 12½-hr cycle.

This scheme shows that a single-basin plant cannot generate power continuously, though it might do so by using a pumped-storage plant if the load it supplies fluctuates sufficiently. A two-basin scheme could generate power continuously without interruption. Tidal plants usually must be of large capacity to make their construction costs economical. None have been built to date.

28-6. Wave Power. The waves of the ocean produced by winds and other causes possess considerable energy. Water waves possess potential energy because of the height difference between crest and trough and also kinetic energy because the particles of water taking part in wave motion move through essentially closed paths. The amount of energy depends on waveshape and the height and distance between crests. Waves 10 ft high and 100 ft between crests have about 30 hp per ft of wavefront.

Many schemes in the past attempted to tap this energy by systems of floats, racks, and gears or by some type of paddle-wheel arrangement. Invariably these mechanical systems could not withstand the pounding and were wrecked in storms.

Figure 28-5 shows an interesting proposal by Arthur D. Little, Inc., to use wave power to generate pneumatic power that could be used for power generation. Waves entering the horn-shaped area between the two jetties become intensified as the passage narrows.

These amplified waves feed energy to a standing wave that fluctuates inside the circular chamber. The inside of the chamber is divided into a smaller circular chamber and an annular chamber formed by a circular wall hung from the ceiling and projecting beneath the water surface. As the standing wave heaves up in the center, it subsides in the annular chamber; this forces air out of the center chamber and induces it into the annular chamber.

As the wave subsides in the center chamber it heaves up at the outer edge to force air out of the annular chamber and induce it into the center chamber. By synchronized valving this air can be made to flow through an air turbine and generate shaft power. A project for a 50-mw plant is being studied.

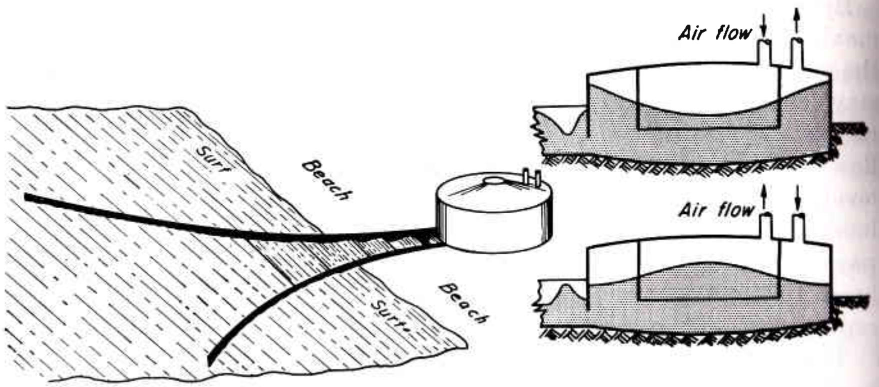


Fig. 28-5. Proposed wave power plant converts wave energy to pneumatic energy in closed chamber. Horn-shaped channel, 1,500 ft long, concentrates waves on 1,000-ft front into compression chamber, 240 ft in diameter and 150 ft high.

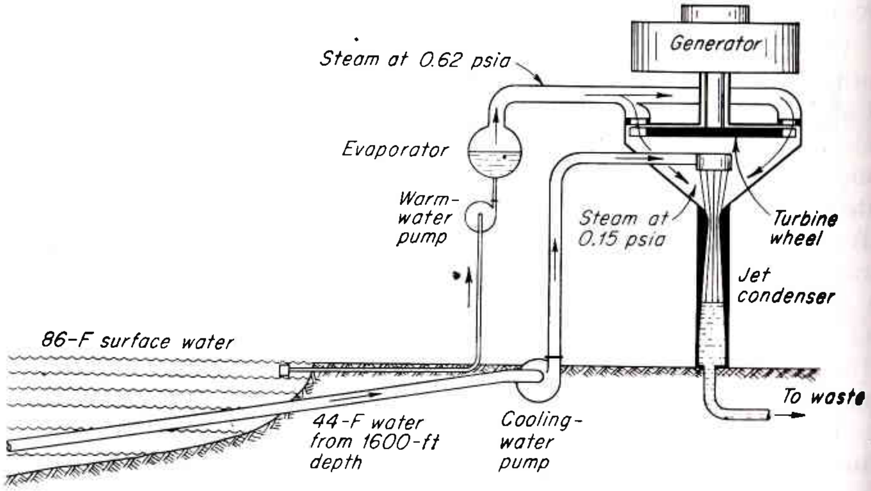


Fig. 28-6. Schematic arrangement of steam power plant working on thermal head available in tropical seas, between 86-F surface water and 44-F bottom water.

28-7. Low-thermal-head Plants.

In tropical seas the surface water may be over 40 F warmer than deeper subsurface waters. The Carnot principle shows that theoretically this thermal difference can produce mechanical power. A Frenchman, Georges Claude, did work on this idea in the 1930s in Cuba, but his experimental plant needed more auxiliary power than it generated.

The French government has since been testing components of a plant to be installed on the west coast of Africa. Figure 28-6 shows a schematic layout of a plant with its principal components. A warm-water pump draws off surface water to be admitted into an evaporator. At 86 F inlet temperature the water evaporates to saturated steam at 0.62 psia. This steam flows to a single-stage turbine and exhausts to a jet condenser at 0.15 psia pressure.

This low pressure is maintained by subsurface water at 44 F, pumped from the depths and used as cooling water. The water jet removes noncondensable gases as well as the exhaust steam. The pressure difference of 0.47 psi can drive the turbine. For the projected plant a turbine with a wheel diameter about 25 ft will generate about 7 mw.

The success of the plant will hinge on keeping the cooling water cold all the way into the condenser and building the suction line (a mile or so long) to keep friction losses low. Pumping energy must be kept low enough to ensure a net plant output. The jet-condenser effluent will have to be discharged far enough away from the plant not to interfere with the surface heating of the ocean water. Tests are also being made of covering the warm water with different types of oil to increase its ability to absorb the heat of solar rays.

28-8. Geothermal Energy.

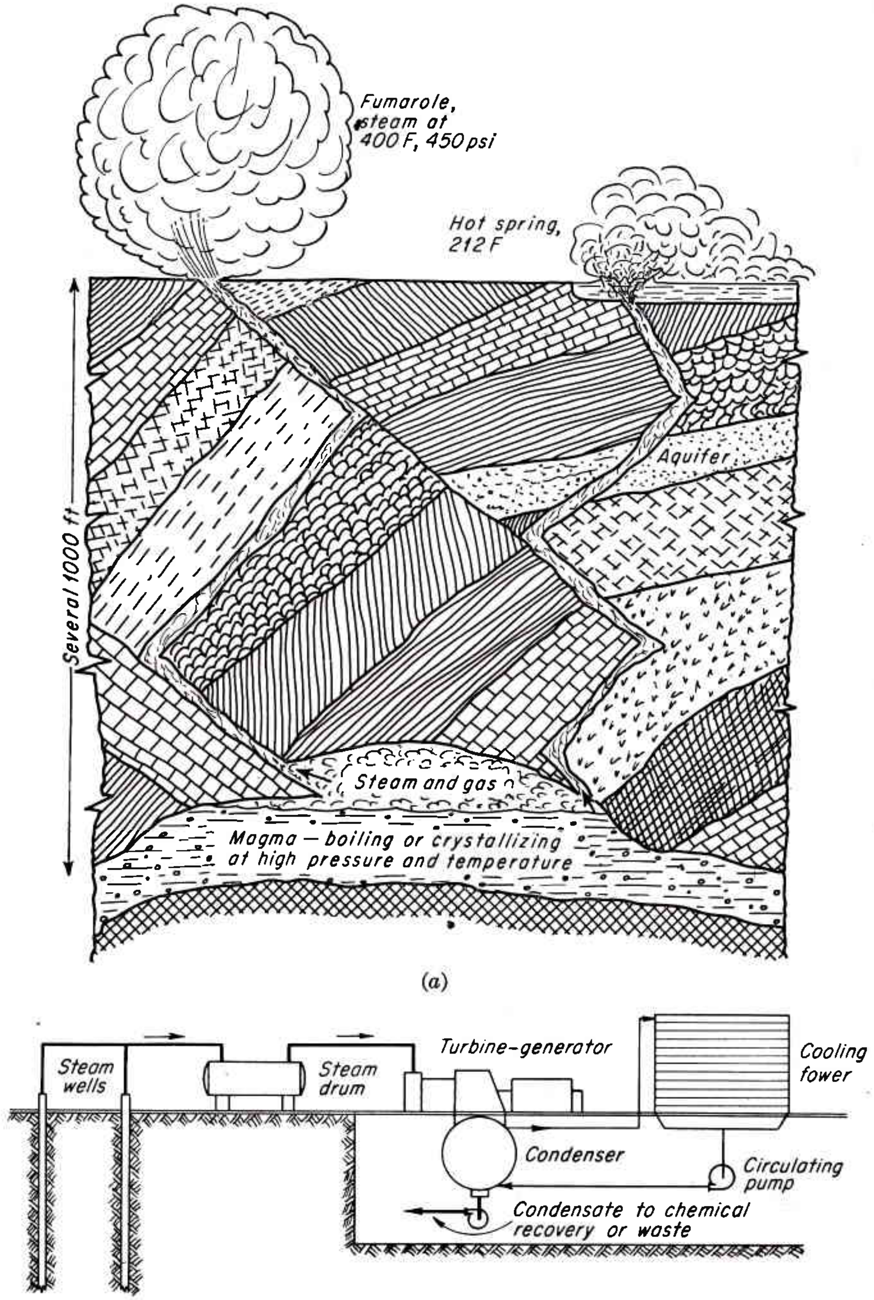
The earth, having been originally thrown off from the sun according to some theories, still has a molten core. Some theorists believe this to be mostly nickel iron. Evidence for this theory lies in the volcanic action that takes place in many regions of the earth's surface. The magma that exudes from the bowels of the earth to form volcanic expulsions also produces fumaroles (steam vents and geysers) and hot springs (Fig. 28-7*a*).

The steam has been used for many years in Italy and Iceland for power generation and space heating. Recent and proposed projects will go into operation in Mexico, New Zealand, the Belgian Congo, and California.

Figure 28-7*b* shows the schematic layout of plants at Lardarello, Italy. Steam is tapped from drilled wells and passes through a drum to separate out moisture. From here the dry steam enters turbines. The steam brings with it valuable chemicals, some of boron base, that are recovered from the entrained moisture and the condensate. Plants in this area have been running since 1910, showing the inexhaustible nature of the energy supply.

28-9. Thermonuclear Energy.

Science works earnestly today at trying to learn methods of fusing the light elements to release nuclear energy. The fusion process releases the enormous amount of energy radiated by our sun and stars. Astrophysicists estimate that the tem-



Fumarole, steam at 400 F, 450 psi
Hot spring, 212 F
Aquifer
Several 1000 ft
Steam and gas
Magma – boiling or crystallizing at high pressure and temperature
Steam wells
Steam drum
Turbine-generator
Cooling tower
Condenser
Circulating pump
Condensate to chemical recovery or waste

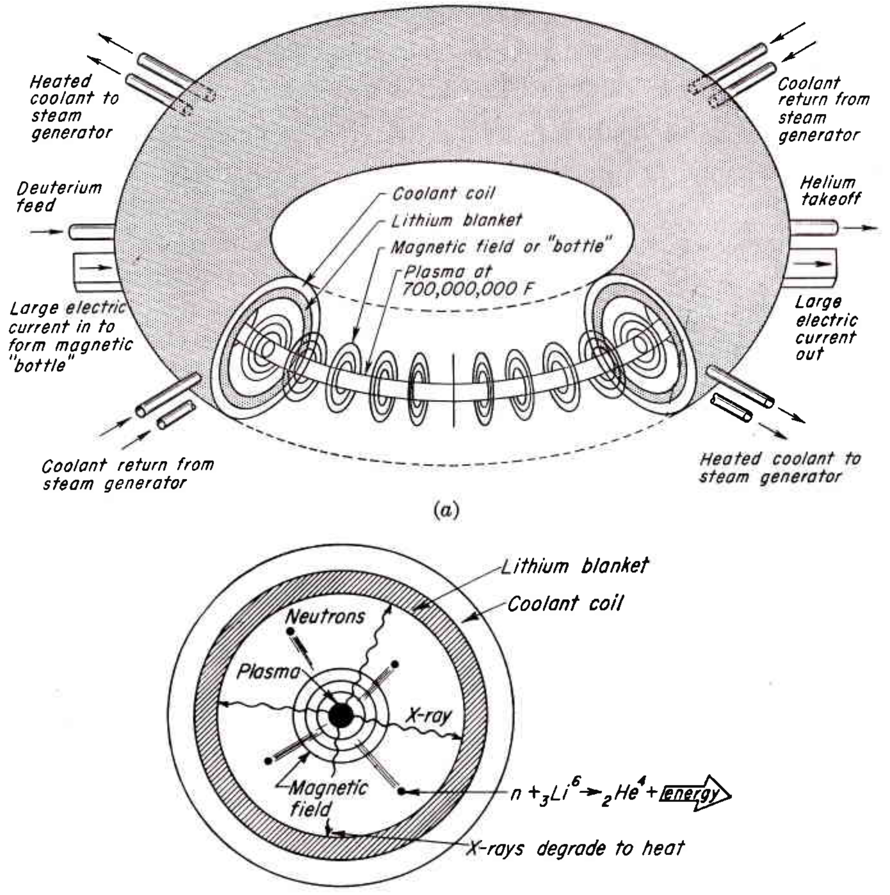
(a)

(b)

FIG. 28-7. (a) Steam vents, geysers, and hot springs receive their heat deep underground from boiling or crystallizing magma; (b) geothermal power plant needs steam drum for separating moisture and solids, turbine generator, main condenser, cooling tower, and their auxiliaries.

perature in the center of our sun is of the order of 30,000,000 F, where the plasma has a density of about eighty-five times that of water.

To run a fusion reactor on earth, our scientists estimate that we shall have to run plasmas at temperatures of about 700,000,000 F and with very low densities. As of this writing, scientists estimate that they have reached temperatures of about 10,000,000 F for durations of about 1 μsec. This compares with an "ignition" temperature of about 100,000,000 F needed by an earth-bound fusion reactor just to start releasing energy.



Heated coolant to steam generator
Deuterium feed
Large electric current in to form magnetic "bottle"
Coolant return from steam generator
Coolant coil
Lithium blanket
Magnetic field or "bottle"
Plasma at 700,000,000 F
Coolant return from steam generator
Helium takeoff
Large electric current out
Heated coolant to steam generator
Lithium blanket
Coolant coil
Neutrons
Plasma
X-ray
Magnetic field
$n + {}_3Li^6 \rightarrow {}_2He^4 +$ energy
X-rays degrade to heat

(a)

(b)

FIG. 28-8. (a) Conjectural arrangement of fusion power reactor to control plasma, using deuterium as fuel and heavy current to form magnetic bottle. Coolant receives heat energy by radiation from plasma. (b) Cross-sectional view.

Fusion reactions would use deuterium (D or H^2) and tritium (T or H^3);

both are isotopes of hydrogen. Deuterium can be found in natural waters, but tritium must be man-made in fusion reactors.

$$_1H^2 + {}_1H^2 = {}_2He^3 + \text{neutron} + 3.3 \text{ mev energy} \qquad (28\text{-}6)$$
$$_1H^2 + {}_1H^2 = {}_1H^3 + {}_1H^1 + 4 \text{ mev energy} \qquad (28\text{-}7)$$
$$_1H^2 + {}_1H^3 = {}_2He^4 + \text{neutron} + 17.6 \text{ mev energy} \qquad (28\text{-}8)$$

where mev is the abbreviation for million electron volts. Energy and tritium may also be reproduced from the reaction

$$_3Li^6 + \text{neutron} = {}_2He^4 + {}_1H^3 + 4.8 \text{ mev energy} \qquad (28\text{-}9)$$

Limited supplies of lithium make deuterium reactions the more likely source of tritium. To put the energy values in scale, note that 1 mev per nucleon = 41.1 × 10^9 Btu per lb. This compares with realizing about 13,000 Btu from the chemical combustion of 1 lb of coal.

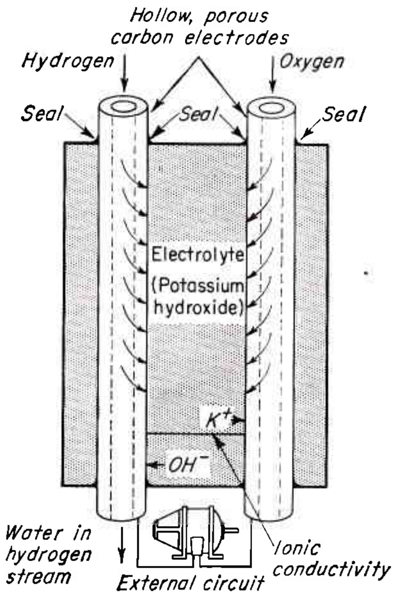


FIG. 28-9. Fuel cell uses hydrogen and oxygen flowing through porous electrodes to react with electrolyte and produce electrical current through outer connected circuit. Cell works continuously as long as hydrogen and oxygen are supplied as fuel.

Fusion reactions take place in a plasma, a mixture of independent light-element nuclei and electrons, a form of ionized gas. Passing an electrical current through the plasma sets up a magnetic field, or "bottle," that compresses the plasma into a small thread and holds it (Fig. 28-8). This *pinch effect* is one method of controlling and heating the plasma so that the nuclei may fuse and release energy. The resistance heating of the electrical current will also raise plasma temperature.

While X rays and gamma rays would be released during the reaction, the wastes of the reaction, largely helium, would not be radioactive, or only weakly so. Several approaches to compressing and heating a plasma are being studied.

28-10. Fuel Cells. These devices convert chemical energy directly to electrical energy. Figure 28-9 shows one form of these cells in schematic form. Hollow, porous, chemically treated carbon electrodes are immersed in a KOH electrolyte confined in a sealed cell.

The electrodes receive the entering hydrogen and oxygen streams and conduct the electrical current produced to the external circuit. Cell

reactions releasing the electrons to the external circuit are essentially

$$O_2 + 2H_2O = 4OH^- - 4e^- \qquad (28\text{-}10)$$
$$2H_2 + 4OH^- = 4H_2O + 4e^- \qquad (28\text{-}11)$$

The entering gases diffuse through the porous carbon to the surface of the electrolyte. At the hydrogen electrode the electrochemical reaction with the electrolyte releases an electron that enters the electrical circuit.

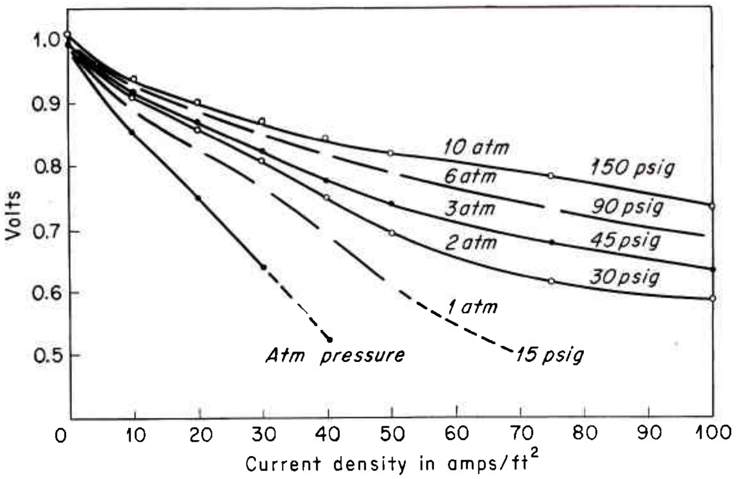


FIG. 28-10. Variation of fuel cell voltage with current density and gas pressures. Cell works at about 150 F temperature.

The electron flows through the external circuit as part of the current and returns to the cell at the oxygen electrode, where the electrochemical reaction of the oxygen and the electrolyte has produced a deficiency of electrons.

A concentric tube cell has been developed with the electrolyte in the annular space. The cell works at ambient temperature and pressure and has continuous life. Conversion efficiencies vary with operation but may be as high as 70 per cent. The conversion efficiency may be estimated by

$$\eta_c = \frac{V}{1.2} \times 100 \qquad (28\text{-}12)$$

where η_c = conversion efficiency, %
V = cell voltage

Cell output can be raised by increasing the gas pressure; higher outputs vary directly with pressure. Figure 28-10 shows the variation of cell voltage with current density and gas pressure. Cells can be connected in series to produce the needed voltage, but the current depends on the physical size of the cell. Fuel cells work best for high-current and low-voltage applications. Work to date indicates an optimum size of about 1 cu ft per kw. Fuel cells have driven a tractor experimentally.

Hydrogen gas may be obtained from generators using calcium or lithium chloride, by catalytic cracking of ammonia or hydrocarbons, or as a by-product from some processes. Oxygen may be used from the air, or from decomposition of peroxides, or pure. About 12.5 cu ft of oxygen and 25 cu ft of hydrogen at standard pressure and temperature are needed per kilowatthour of energy.

28-11. Thermoelectric Power. In 1822 the German physicist Seebeck discovered that a loop of two dissimilar metals developed an emf when the two junctions were kept at different temperatures. This effect has long been used in thermocouples to measure temperatures. This phenomenon offers one method of producing electrical energy directly from the heat of combustion, but its thermal efficiency is of the order of 1 to 3 per cent, not very attractive except for very special small power applications.

The basic simplicity of this method of electric-power production has encouraged research to find materials that might work more efficiently. Much of the study has centered on the semiconductor class of materials that stand midway between electrical insulators and conductors. The material in addition to working with reasonable efficiency must also be able to withstand high-temperature service. Encouraging results have been obtained with ceramic-type materials made of semiconductors "doped" with materials to increase their conductivity to a level about halfway between semiconductors and metal conductors.

These materials, known as mixed-valence metals, include materials such as manganese, iron, cobalt and nickel treated with oxygen, sulfur, selinium, and tellurium. Researchers report efficiencies of the order of 8 per cent and confidently expect to develop materials that would work at 20 per cent thermal efficiency. They envision the probability of developing peak-load power stations with capacities of the order of 100,000 kw. If these stations can be built at a low capital investment, they would be highly acceptable for that duty as well as for emergency stand-by service or even for base load service, where fuels are very cheap.

28-12. Thermoionic Converter. Another form of direct conversion of heat energy to electrical energy has been achieved in the thermoionic converter (Fig. 28-11). This consists of two electrodes held in a container filled with ionized cesium vapor. Heating one electrode "boils out" electrons that travel to the opposite, colder electrode. The positive ions in the gas neutralize the space-charge effect of the electrons that normally impedes electron travel through a gas. Connecting the two electrodes through an external circuit permits an electrical current to flow.

Reported efficiencies of these types of tubes is of the order of 8 per cent, but researchers expect to produce units working at thermal efficiencies of 30 per cent. At this efficiency the emitting cathode will be sized at 100 sq cm for a 1-kw output. Potential will be d-c at about 2 to 3 volts. The present cathode is heated to 2500 F. Theoretically these units could be connected in series or parallel for different plant voltages and capacities.

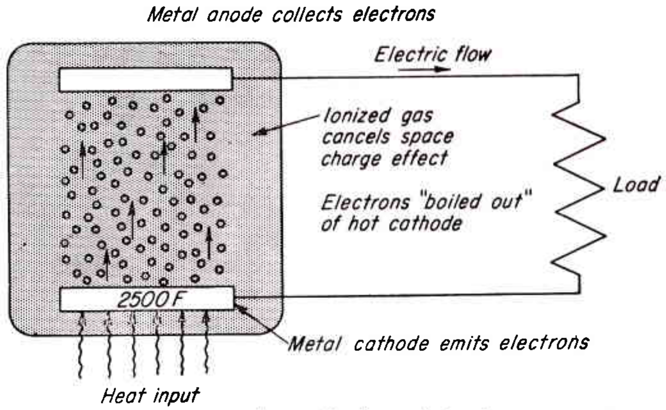

Metal anode collects electrons
Electric flow
Ionized gas cancels space charge effect
Electrons "boiled out" of hot cathode
Load
2500 F
Metal cathode emits electrons
Heat input

FIG. 28-11. Thermoionic converter has cathode receiving heat that boils out electrons. These migrate to cooler anode collector and flow through outer circuit to develop electric power. Ionized gas offsets space-charge effect that tends to repel migration of electrons.

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PART V

POWER ECONOMICS

CHAPTER 29

THE GENERAL ECONOMIC PROBLEM

29-1. Economics. Engineering thermodynamics emphasizes the maximum efficiency that can be theoretically obtained for given conditions in a power-producing cycle. Unfortunately this emphasis often gives the student a strong bias in favor of the most efficient equipment in his attempts to apply theoretical principles to practical conditions. From the broadest viewpoint a power-producing system is only an accessory device needed to serve some human need such as the manufacture of an article or the supply of light and heat or to enable the functioning of transportation and communication systems. These ultimate objectives the designer must strive to serve with maximum effectiveness. The effectiveness is measured financially. When the *total over-all* monetary cost of energy service required to meet the ultimate needs has been brought down to an irreducible minimum for the particular circumstances, the power-system designer has fully performed his function. The thermal efficiency of a plant is only one factor of many that determine the final energy cost. In the vast majority of cases the *most efficient* plant thermally is *not* the economic one.

As in all engineering economics, the principles of power-system economics use certain well-established techniques in making the most suitable single selection of a host of available alternatives. Although only the supply of electrical energy will be considered in this treatment, keep in mind that the basic methods can be applied to any of the other forms of energy conversion and distribution systems, such as manufactured gas, steam, compressed air, and refrigeration.

29-2. Cost of Service. Energy-supply problems may be broadly classified as follows: (1) erection of an entirely new energy system, (2) replacement of an existing energy system, and (3) extension of an existing energy system. A variety of physical ways may satisfy each type of requirement, and the designer must select the most economic one. In studying each solution of a particular problem the various costs incurred must be estimated. Initially there will be the cost covering the purchase, installation, and erection of the power-system equipment. This cost, incurred only once during the life of a simple project, is often referred to as the capital of the enterprise and in turn leads to costs that are incurred annually during the life of the project. These latter costs are termed *investment charges*, or *fixed charges* and consist of interest, taxes, insurance, and depreciation or amortization. The individual fixed charges may remain substantially constant from year to year.

In addition to the above, annual operating costs must be met. These cover expenditures for fuel, labor, repairs, supplies, supervision, and certain taxes. The operating costs of a system are generally variable in magnitude and depend on the amount of energy produced.

These costs of diverse natures must be reduced to a basis upon which they can be easily compared for the various alternatives. Several such bases are in common use, the one selected depending (1) on the behavior of the periodic costs during the life of the project and (2) on the ease of interpreting results. These methods of comparison will be discussed after the nature of the individual costs has been studied.

INVESTMENT CHARGES, OR FIXED CHARGES

29-3. Cost of Money Use—Interest. When a person lends some of his funds to a borrower for the latter's use, he deprives himself of the opportunity to apply those funds in satisfying an immediate personal want. Furthermore, the lender risks the safety of the funds since he cannot supervise their application by the borrower. Some inducement must be offered the lender by the borrower to compensate for these two inconveniences. This inducement usually is made by returning to the lender a greater amount than was originally loaned.

The difference between the amount returned and the loan is the *interest* or the *cost of money use* to the borrower. Payment of interest by the borrower is possible only by using the funds in some productive enterprise, i.e., the combination of the borrower's efforts and initiative plus the borrowed funds creates value in some commodity or service. Selling this commodity or service enables the borrower to pay all his costs, recover the borrowed funds, and in addition pay the interest on them.

Interest is expressed in terms of a rate, i.e., an interest rate. A 12-month period is the common time basis. Thus if \$100 is borrowed at the beginning of a 12-month period and \$105 is returned at the expiration of the period, the interest is \$5 and the interest rate is $\$5/\$100 = 0.05$, or 5 per cent per annum. Normally the expression "per annum" is omitted, and any interest rate expressed simply as a percentage is understood to apply to a 12-month period.

29-4. Interest Applications. There are several ways of paying interest on a loan that extends over a period of years. Necessarily the interest payments are closely associated with the scheme adopted to repay the original loan. These plans are best illustrated by numerical examples. Consider a loan of \$100 made for a 5-year period at 4 per cent interest rate. The loan may variously be termed the *principal* or *investment*. When the loan is repaid only at the expiration of the 5-year period, two methods of interest payment may be adopted. The simpler requires payment of the \$4 interest to the lender at the end of each year. Table 29-1 shows the schedule of payments over the life of the loan for this scheme. Under this schedule the lender receiving the \$4 interest payment may invest this sum in some other enterprise and in turn earn interest or a return on it.

TABLE 29-1. SCHEDULE OF PAYMENTS ON \$100 LOAN AT 4 PER CENT INTEREST FOR 5 YEARS. NO PERIODIC REPAYMENTS OF LOAN. ANNUAL INTEREST PAYMENTS

End of year	Amount of loan owed	Repayment on loan	Payment of interest	Total year-end payment
0	\$100			
1	100	0	\$4	\$4
2	100	0	4	4
3	100	0	4	4
4	100	0	4	4
5	100	\$100	4	104
Total.	. . . .	\$100	\$20	\$120

In lieu of this method the borrower may be permitted to retain the interest to use in his enterprise and earn additional interest thereon. At the end of the loan period the borrower pays one lump sum to the lender, including the original sum borrowed, the annual interest earned on the original loan, and the interest earned on the annual interest earned. This method of placing a loan is commonly termed *investing at compound interest*. Table 29-2 illustrates the schedule of payments of \$100 lent for a 5-year period at 4 per cent interest, compounded annually.

A comparison of Table 29-1 with Table 29-2 shows that the interest actually paid out by the borrower to the lender is \$1.67 more for the compound-interest scheme over the life of the loan. This represents the earnings of the annual interest retained in the borrower's business. From the lender's viewpoint either scheme will yield the same total earnings since, theoretically at least, he can reinvest the annual interest payments to earn the increment of \$1.67. From the borrower's viewpoint he can accept the compound-interest plan only if his enterprise is capable of expansion in which the annual interest can be invested. If such expansion is not inherent in the business, he will elect to pay the interest annually.

Frequently part of the original loan is repaid periodically so as to distribute the amount as a more or less uniform annual expense. This periodic repayment of the original loan or principal is commonly termed the "amortization" of the loan or principal. The two most common schemes are calculated (1) to pay equal parts of the loan each year or period with varying total interest payments or (2) to pay equal sums for each year of the life of the loan so that at the expiration date all interest and the original loan or principal have been paid back. Table 29-3 illustrates the first scheme using uniform amortization payments of a loan of $100 at 4 per cent interest for a period of 5 years.

TABLE 29-2. SCHEDULE OF PAYMENTS ON $100 LOAN AT 4 PER CENT INTEREST COMPOUNDED ANNUALLY FOR 5 YEARS. NO PERIODIC REPAYMENTS OF LOAN. NO ANNUAL INTEREST PAYMENTS

End of year	Amount of loan owed	Repayment on loan	Payment of interest	Total year-end payment
0	$100.00			
1	104.00	0	0	0
2	108.16	0	0	0
3	112.49	0	0	0
4	116.99	0	0	0
5	121.67	$100	$21.67	$121.67
Total........	....	$100	$21.67	$121.67

TABLE 29-3. SCHEDULE OF PAYMENTS ON $100 LOAN AT 4 PER CENT INTEREST FOR 5 YEARS. EQUAL PERIODIC REPAYMENTS OF LOAN. VARYING ANNUAL INTEREST PAYMENTS

End of year	Amount of loan owed	Repayment on loan	Payment of interest	Total year-end payment
0	$100			
1	100	$ 20	$ 4.00	$ 24.00
2	80	20	3.20	23.20
3	60	20	2.40	22.40
4	40	20	1.60	21.60
5	20	20	0.80	20.80
Total........	....	$100	$12.00	$112.00

A comparison of the three tables will show that the interest payments are a minimum for the last. This is readily explained by the reduction of the outstanding unpaid balance of the loan as time progresses.

Table 29-4 illustrates the schedule for the second scheme of amortization resulting in uniform total annual payments. In Table 29-4 interest payments exceed those of Table 29-3 by 30 cents. This is caused by the less rapid reduction of outstanding unpaid balance of the loan as time progresses. Although the borrower makes a greater total repayment over the life of the loan by making uniform total year-end payments, this disadvantage is offset by the convenience of making smaller payments in the early life of the loan.

TABLE 29-4. SCHEDULE OF PAYMENTS ON $100 LOAN AT 4 PER CENT INTEREST FOR 5 YEARS. VARYING PERIODIC REPAYMENTS OF LOAN. VARYING ANNUAL INTEREST PAYMENTS. UNIFORM TOTAL ANNUAL PAYMENTS

End of year	Amount of loan owed	Repayment on loan	Payment of interest	Total year-end payment
0	$100.00			
1	100.00	$ 18.46	$ 4.00	$ 22.46
2	81.54	19.20	3.26	22.46
3	62.34	19.97	2.49	22.46
4	42.37	20.77	1.69	22.46
5	21.60	21.60	0.86	22.46
Total	....	$100.00	$12.30	$112.30

In these tables four different schedules of total year-end payments are shown for a loan or *principal* of $100, an interest rate of 4 per cent, and for the period of 5 years. In any of the schedules it may be said that they have a present worth of $100 at 4 per cent interest. This concept of present worth of a uniform series of payments, a varying series of payments, or lump-sum payment in the future all being equal at a given interest rate is an important tool in making economic comparisons.

29-5. Financial Mathematics. Although in most practical business dealings interest is paid annually or oftener, the concept of compound interest, i.e., reinvesting interest to earn interest, is of great importance in economic evaluations. Standard formulas for compound-interest calculations have been developed and are in common use. With given interest rates and loan periods we often wish to determine: (1) the sum of money accumulated at the end of the period for a given sum invested now; (2) the inverse condition, the present value or worth of a sum available at some future date; (3) the accumulation of a series of periodic payments or investments; (4) the inverse condition, the periodic payments required to provide a given future sum; (5) the annual payments, or annuities, that will be received for a given sum of money invested now; and (6) the inverse condition, the present worth of a future schedule of payments.

To determine the sum accumulated at the end of a given period of years for a given sum invested now at interest compounded annually, let

P = principal, investment, or amount of loan at beginning of period, dollars
S = accumulated sum at end of period, dollars
i = annual interest rate, expressed as a decimal
n = length of period, years

Then, at the end of first year,

$$S_1 = P + iP = P(1 + i)$$

at the end of second year,

$$S_2 = P(1 + i) + iP(1 + i) = P(1 + i)^2$$

at the end of third year,

$$S_3 = P(1 + i)^2 + iP(1 + i)^2 = P(1 + i)^3$$

From this it becomes evident that, at the end of the nth year,

$$S = P(1 + i)^n *$$

It follows that

$$P = \frac{S}{(1 + i)^n}$$

Example 29-1. What will be repaid at the end of 15 years to the person who lends $120 at 4 per cent interest compounded annually?

Solution

$$\begin{aligned} S &= P(1 + i)^n \\ &= 120 \times 1.04^{15} \\ &= 120 \times 1.801 = \$216.12 \end{aligned}$$

Example 29-2. What is the present worth of receiving $1,000 twenty years from now if money can earn 5 per cent interest compounded annually?

Solution

$$\begin{aligned} P &= \frac{S}{(1 + i)^n} \\ &= \frac{1{,}000}{1.05^{20}} \\ &= \frac{1{,}000}{2.653} = \$376.93 \end{aligned}$$

Interest is often paid or compounded more frequently than once a year. If m is the number of times a year that interest is paid, then

$$S = P\left(1 + \frac{i}{m}\right)^{mn}$$

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* For tabular values of $(1 + i)^n$, see Appendix Table 14.

Example 29-3. Find the accumulation of $100 invested for 20 years at 6 per cent interest if:

a. Interest is compounded annually.
b. Interest is compounded semiannually.
c. Interest is compounded monthly.

Solution

$$S = P\left(1 + \frac{i}{m}\right)^{mn}$$

$$\begin{aligned} a.\quad S &= 100\left(1 + \frac{0.06}{1}\right)^{1\times 20} \\ &= 100 \times 1.06^{20} = \$320.70 \\ b.\quad S &= 100\left(1 + \frac{0.06}{2}\right)^{2\times 20} \\ &= 100 \times 1.03^{40} = \$326.20 \\ c.\quad S &= 100\left(1 + \frac{0.06}{12}\right)^{12\times 20} \\ &= 100 \times 1.005^{240} = \$331.00 \end{aligned}$$

To determine the sum accumulated at the end of a given period of years for a series of loans or investments of equal magnitude made annually, let

A = investment made at *end* of each year for n years

Then, accumulation of first year's investment is

$$S_1 = A(1 + i)^{n-1}$$

accumulation of second year's investment is

$$S_2 = A(1 + i)^{n-2}$$

accumulation of third year's investment is

$$S_3 = A(1 + i)^{n-3}$$

and so forth, till $n - 2$ years' investment is

$$S_{n-2} = A(1 + i)^2$$

accumulation of $n - 1$ years' investment is

$$S_{n-1} = A(1 + i)$$

accumulation of nth years' investment is

$$S_n = A$$

and total accumulation is

$$S = A[1 + (1 + i) + (1 + i)^2 + \cdots + (1 + i)^{n-2} + (1 + i)^{n-1}]$$

Multiplying both sides by $1 + i$,

$$(1 + i)S = A[(1 + i) + (1 + i)^2 + (1 + i)^3 + \cdots + (1 + i)^{n-1} + (1 + i)^n]$$

Subtracting the two equations from each other,

$$iS = A[(1 + i)^n - 1]$$

Then

$$S = A\,\frac{(1 + i)^n - 1}{i}$$

By transposing this equation, the annuity A required to yield a given sum S in a sinking fund is

anualidade de capitalização

$$A = S\left[\frac{i}{(1 + i)^n - 1}\right]^*$$

From the relationship of the principal P and the accumulation S the present worth of a series of constant annuities A may be readily established as follows:

$$A = S\,\frac{i}{(1 + i)^n - 1} = P(1 + i)^n\,\frac{i}{(1 + i)^n - 1}$$

Anualidade de amortização

$$= P\left[\frac{i}{1 - (1 + i)^{-n}}\right]^\dagger = \frac{(1+i)^n \cdot i}{(1+i)^n - 1} = A_c(1+i)^n$$

which may be interpreted as the constant annuity that may be purchased for n years by investing P now at i interest rate. Then the present worth of the series of annuities will be

$$P = A\,\frac{1 - (1 + i)^{-n}}{i}$$

An important identity that should be noted is:

$$A = P\left[\frac{i}{1 - (1 + i)^{-n}}\right] = P\left[\frac{i}{(1 + i)^n - 1} + i\right]$$

That is, the two bracketed terms are equal, the right-hand term being the sum of the fractional coefficient used to determine A for a given value of S and the interest rate. In other words, the annuity A is the annual year-end payment that will repay the total amount P originally loaned, plus the interest earned thereon.

Example 29-4. How much must be deposited at the end of each year in a sinking fund for 15 years to accumulate \$1,000 if interest is at 5 per cent?

Solution

$$\begin{aligned} A &= S\,\frac{i}{(1 + i)^n - 1} \\ &= 1{,}000 \times \frac{0.05}{1.05^{15} - 1} \\ &= 1{,}000 \times 0.04634 = \$46.34 \end{aligned}$$

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* For tabular values of $i/[(1 + i)^n - 1]$, see Appendix Table 15.

† For tabular values of $i/[1 - (1 + i)^{-n}]$, see Appendix Table 16.

Example 29-5. If \$50 were deposited at the end of each year for 12 years in a sinking fund earning 6 per cent, what would be the total accumulation?

Solution

$$\begin{aligned} S &= A\,\frac{(1 + i)^n - 1}{i} \\ &= 50 \times \frac{1.06^{12} - 1}{0.06} \\ &= 50 \times 16.870 = \$843.50 \end{aligned}$$

Example 29-6. Find the uniform year-end payment that will repay a present loan of \$500 at an interest rate of 4 per cent in 20 years.

Solution

$$\begin{aligned} A &= P\,\frac{i}{1 - (1 + i)^{-n}} \\ &= 500 \times \frac{0.04}{1 - 1.04^{-20}} \\ &= 500 \times 0.07358 = \$36.79 \end{aligned}$$

Example 29-7. What is the present worth of \$200 received at the end of each year for 10 years if money earns 3 per cent interest?

Solution

$$\begin{aligned} P &= A\,\frac{1 - (1 + i)^{-n}}{i} \\ &= 200 \times \frac{1 - 1.03^{-10}}{0.03} \\ &= 200 \times 8.530 = \$1{,}706 \end{aligned}$$

29-6. Nominal and Effective Interest Rates. Corporate bonds are usually issued at face values of \$1,000 with a stated nominal interest rate based on the face value. Coupons are attached to the bond and, upon presentation to the issuing corporation, will be honored by payment of interest due for the current period. Usually the interest is payable semiannually. Although the value of the interest is fixed, the rate of interest that the purchaser receives is usually different from the nominal rate, whereas the rate of interest the seller pays is usually higher than either of the former, as demonstrated in the following discussion.

If a \$1,000 bond carries coupons valued at \$20, each payable semiannually for a period of 10 years, the nominal interest rate is 4 per cent. However, in floating the bond issue the corporation usually receives less than the face value from the bankers who market the securities. The investment bankers in turn sell the bonds to the public at a price higher than they paid the issuing corporation and so cover their costs and gain a profit. Depending upon the availability of investment funds, the final selling price of the bonds may be more or less than the face value, usually less. Besides selling the bonds below face value, a corporation incurs expenses in preparing the bond issue such as engraving, accounting, registration, and legal fees. Furthermore the mechanics of paying the semiannual interest incur periodic expenses which further reduce the net amount that may be employed directly in the corporation's production activity.

Thus, in the bond mentioned, the price paid by the investment bankers may have been $920 and they in turn sold it for $960. The initial selling expense of the corporation may have been $20 so that it nets $900 for investment in its production activity. In addition, the bookkeeping and distributing cost of paying the interest may be as much as $2 per coupon. From the purchaser's viewpoint he has paid $960 for a promise to receive $20 every 6 months for 10 years and $1,000 at the end of 10 years. From the corporation's viewpoint it has received $900 for which it promises to pay $22 every 6 months for 10 years and $1,000 at the expiration of that period.

To determine the interest rate for each party to the bond, it is necessary to set up the present-worth equation and solve for i by trial and error. Thus, for the final purchaser of the bond,

$$P = A\frac{1 - (1 + i)^{-n}}{i} + \frac{S}{(1 + i)^n}$$

where the interest rate per period is greater than 2 per cent and there are 20 periods, then,

$$960 = 20 \times \frac{1 - (1 + i)^{-20}}{i} + \frac{1,000}{(1 + i)^{20}}$$

Solving,

$$i = 0.0226, \text{ or } 2.26\%, \text{ every 6 months, or } 4.52\% \text{ annually}$$

That is, the purchaser receives 4.52 per cent interest for his investment though the nominal coupon rate is 4 per cent.

Examining the cost of the money to the corporation,

$$900 = 22 \times \frac{1 - (1 + i)^{-20}}{i} + \frac{1,000}{(1 + i)^{20}}$$

Solving,

$$i = 0.0286, \text{ or } 2.86\%, \text{ every 6 months, or } 5.72\% \text{ annually}$$

The great difference that may exist in the interest rate becomes readily apparent. Though both parties are concerned with the same security having a face-value interest rate of 4 per cent, the lender earns 4.52 per cent and the borrower spends 5.72 per cent.

29-7. Investment. An initial capital outlay must be made to buy the land and equipment comprising an energy-supply system as well as to assemble and install it ready for use. The money for this may be acquired by the sale of stocks or bonds or both; or, in a private enterprise, it may be taken from the personal funds of the owner.

The sale of stocks in a business venture implies the sale of a share of ownership in the enterprise. Such diverse ownership shares, in proportion to the money paid for the stocks, in management, profit, and loss of the business. Stocks can be sold if the prospects for future annual returns or dividends are good and the integrity of the promoters of the enterprise is established. Usually the dividend must be at least equal to the best rate of return that can be obtained by lending the money (as by buying bonds) plus an additional amount to compensate for the risks of ownership.

A bond sold by a business venture is a means of borrowing money by pledging the physical and financial assets of the venture as collateral or security. Thus the bondholders have first claim upon the earnings of a business. In years of high business activity this usually means that high dividends can be distributed to the stockholders; in times of low business activity the interest due the bondholders and the associated expense may absorb all the earnings and leave none for the stockholders. If for any extended period the earnings are insufficient to cover the bond interest, the business usually fails.

In supplying energy either for a utility or for a private industrial enterprise the bond interest and dividends must be accounted for as part of the total cost of service. The situation is not different when the money to build a private power system is supplied from the personal funds of the owner. Theoretically the owner can invest the money used for construction of the plant in some other enterprise and earn a yield on such investment. It follows that the plant must show prospects of yielding an equivalent return or else its construction should not be undertaken.

29-8. Depreciation and Amortization. In general, man-made devices are impermanent; stated in another fashion, man-made devices depreciate. This deterioration, or depreciation, of equipment takes place as time proceeds. At any instant it is very difficult to determine the exact rate of depreciation, or the total amount of depreciation since installation of the equipment. Many factors influence the depreciation of capital equipment, but in general they fall into five main categories: (1) life of the enterprise, (2) life of the equipment, (3) inadequacy of equipment, (4) obsolescence of equipment, and (5) requirements of public authority.

1. In some instances the life of an enterprise may be curtailed, as when the demand for its products vanishes. Cessation of the demand may or may not be foreseen. In this case the capital equipment may be in excellent physical condition for the purpose for which it was needed.

If it can be adapted to serve other uses, it may be readily sold and thereby have a salvage value. If it cannot be adapted to other uses, its only value may be as scrap materials with a low salvage value. It has happened that equipment with no salvage value incurs an expense in removing it from its location.

2. From the moment man-made devices are first completely installed they steadily deteriorate from corrosion, rot, and weathering as well as from the wear and tear of use. Depreciation from these causes can be remedied often by repairs. By a thorough program of preventive maintenance it is possible to keep capital equipment equal to new, i.e., the depreciation to zero. Since various parts of a plant will have different rates of deterioration, the question of the rate of deterioration is complicated. Most authorities seem to agree that for economy studies the probable life of capital equipment is generally longer than should be assumed in placing the conclusions of such studies on a conservative basis. Where the life of equipment is taken to be indeterminate, some engineering organizations assume that the salvage value will just equal the cost of removal. This is a simplifying assumption and is justified in view of the uncertainties.

3. Capital equipment becomes inadequate when the demands upon it exceed its capacity to produce. In this case new equipment may be added to aid in carrying the increased demands, or it may be necessary to remove the existing equipment and replace it with new and larger equipment capable of satisfying the new demands. This latter condition has frequently occurred in the history of power generation.

4. A plant becomes obsolete when it can be replaced by one of more modern design which operates at a reduction in total annual costs. The reduction of costs may come about from better efficiency in the replacement plant or from deterioration of performance of the existing plant. For the former condition the obsolescence of an existing plant can be determined only by comparing its operating cost with an estimate of the total annual costs of a new plant, when such a new plant of improved design actually becomes available. If the performance of the existing plant is unaffected by use, then for a rapid rate of technological development the plant may possibly become obsolete in a very short time. This possibility of early obsolescence is often one of the chief reasons for assuming shorter lives for projects than the expected physical lives of the equipment involved.

Regardless of the causes of depreciation it is evident that the capital of an enterprise must remain intact for financial stability and safety. When a plant becomes useless for any cause, funds should be in hand to build a new one, if circumstances warrant, or to reimburse the original sources of the capital. Ideally, at any instant during the life of an enterprise, the depreciated value of the plant plus the accumulation in the depreciation fund should equal the original invested capital. This is seldom attainable in practice. The major objective must still be fulfilled, however, and that is the amortization of the invested capital funds. In practice the rate of amortization and the rate of physical depreciation are seldom precisely correlated. In accounting practice several methods are available to charge income with amortization of the capital funds or, as commonly stated, to allow for depreciation. These are usually designated as: (1) straight-line method, (2) percentage method, (3) sinking-fund method, (4) appraisal method, and (5) unit method.

1. In the straight-line method the life of the enterprise or of the equipment (whichever is shorter) is estimated. The capital cost less the salvage value of the equipment is divided by the number of years of life, and this amount is deducted from income annually as amortization of capital funds. This method is the most common in application, principally because of its simplicity.

2. The percentage method differs from the straight-line method in that, instead of uniform depreciation charges for all years, the depreciation charges vary from year to year. This method assumes that the value of the capital equipment decreases by a constant percentage from its value of the previous year; the succeeding annual depreciation charges diminish progressively but never reach zero. Of course, the remaining capital value will reach an estimated salvage value. Included in this method are arbitrary tables of percentages that will vary the magnitude of the annual depreciation charges so that they will either progressively increase or diminish.

3. The sinking-fund method is based on the conception that the annual uniform deduction from income for depreciation will, when invested at a given interest rate, accumulate to the capital value of the enterprise at the termination of the venture, or the end of life for the equipment, as the case may be. For this method, although the annual allowance is uniform it will be appreciably less than that calculated by the straight-line method. Though this method does not find very frequent application in practical depreciation accounting, it is the fundamental method used in making economy studies. This follows from the basic fact that any sums realized as amortization of the capital funds can (theoretically) be reinvested and yield a return.

4. At the end of each year an appraisal is made of the value of the capital equipment, the difference in annual appraisals being then taken as the depreciation for the intervening year. This method is costly and impractical for properties of any magnitude and is not often applied.

5. In the unit method some factor is taken as a standard and the depreciation measured thereby. Thus the total number of working

hours that the equipment or enterprise may be capable of during its entire life is first estimated. Dividing the capital funds by the number of hours will give the amount of capital consumed or depreciated per working hour. Then each year this constant is multiplied by the number of actual working hours to arrive at the magnitude of depreciation for that year. Instead of working hours, the number of units of production may be used as the measuring standard.

29-9. Disposal of Depreciation Charges. In the accounting procedure, depreciation is deducted from income and allocated to depreciation reserve. Depreciation reserve is merely an account that reflects the estimated decrease in value of capital equipment and does not necessarily represent a concrete sum of money set aside and available on call. As these annual sums are taken out of income or revenue, they are recorded in the depreciation reserve but must be disposed of either by retaining them in the enterprise or by reimbursing the sources of the capital funds.

These sums are retained in the business when invested in capital equipment either by expansion of the plant or by replacement of worn-out equipment. This is the usual procedure if the business has any possibility of continuing indefinitely.

Where capital funds have been obtained by bond issues, it is sometimes required to establish sinking funds to guarantee repayment of principal of the bond at the time of maturity. Thus part of the depreciation charges would be so invested annually. Another method of repaying borrowed funds has serially maturing bonds. In this procedure a given number of bonds determined by lot are redeemed at face value each year until the entire issue has been retired. The bonds thus repurchased represent the depreciation reserve.

29-10. Taxes. Taxes levied on an enterprise are many and quite diverse as to the basis used for computing them. The land upon which a plant stands is usually taxed at a rate depending upon its assessed valuation, which is determined from its location with regard to certain natural and cultural advantages. The tax levied on the value of the capital equipment comprising the plant is usually known as a *property* tax. Franchise taxes borne by a public-utility system in theory pay for the use of the public streets by its equipment. There are numerous other forms of taxes such as social security, unemployment, income, excess profits, and sales. Some taxes depend entirely upon the magnitude of the capital funds, others on the volume of business, and still others on a combination of the two factors. Taxes that are a function of capital investment only should be included in the investment charges; other taxes should be charged to operating costs.

Though there are numerous forms of taxes, their cost is often multiplied by the existence of several taxing authorities, many of which will levy the same type of tax on an enterprise. Care should be used to satisfy the tax demands of town, county, state, and Federal authorities.

29-11. Insurance. Every well-managed company carries some forms of insurance against accidents to equipment and personnel. Since the risks involved are diverse in nature, an insurance list is quite long. On it will be found fire, windstorm, hail, flood, earthquake, explosion, loss of use, public liability, workmen's compensation, automobile, marine, title, fidelity, forgery, credit, and many others. Insurance is usually carried by conventional insurance companies, but if an enterprise is very large and has its properties scattered over a considerable region, it may often be self-insured. This is done by setting aside a reserve fund built up periodically out of earnings. An insurance fund of this type should not be confused with a depreciation fund.

29-12. Total Fixed Charges. Initial expenditure of funds for capital equipment incurs annual costs that must be met out of income. These annual costs, for an enterprise that is not subject to expansion, remain largely constant from year to year, though they are subject to variations in tax and insurance rates. The annual *investment charges*, or as they are more commonly termed the *annual fixed costs*, are usually expressed in terms of a percentage of the capital cost. These percentages must be determined individually, as two cases are seldom alike. For any single company these percentages or fixed-charge rates must be redetermined from time to time as the various component cost rates fluctuate or as the possibilities of future changes in these rates present themselves.

The component of the fixed-charge rate that represents the cost of money use depends on the proportion of capital funds obtained from bond and stock issues. An example will show how this component is computed.

Example 29-8. Find the cost of money use for a business that has obtained 40 per cent of its capital funds by a bond issue that costs it 4.6 per cent as an interest rate payable every 6 months. The remaining 60 per cent of capital has been raised by a stock issue on which earnings of 6 per cent payable annually are expected.

Solution. For every dollar of invested capital 40 cents must earn 2.3 per cent every 6 months and the remaining 60 cents earn 6 per cent every 12 months. The annual earnings of the 40-cent portion are

$$\begin{aligned} S - P &= P(1 + i)^n - P \\ &= 0.4 \times 1.023^2 - 0.4 = 0.4 \times 0.0465 \\ &= 0.0186, \text{ or } 1.86 \text{ cents per year} \end{aligned}$$

The annual earnings of the 60-cent portion are

$$= iP = 0.06 \times 0.60 = 0.036, \text{ or } 3.6 \text{ cents per year}$$

Then the total earnings that must be realized per dollar are

$$= 1.86 + 3.60 = 5.46 \text{ cents}$$

or the interest rate that must be realized is 5.46 per cent.

The ratio of the annual depreciation to the capital invested is the depreciation rate. From the formulations already developed under Sec. 29-5, Financial Mathematics, as the sinking-fund method this is

$$\frac{A}{S} = \frac{i}{(1+i)^n - 1}$$

where A = annual depreciation charged against annual income, dollars
S = value of capital funds that must be accumulated, dollars
n = expected life of capital equipment or business, years
i = earnings rate or interest that investment of A will yield

Example 29-9. Find the depreciation rate for a business that is expanding and is earning 5 per cent on its invested capital if the expected life of the equipment is 15 years.

Solution

$$\frac{A}{S} = \frac{i}{(1+i)^n - 1} = \frac{0.05}{1.05^{15} - 1} = 0.0463$$

Or, if at the end of each year an amount equal to 4.63 per cent of the investment is reinvested in the business, the original capital will be recovered at the time the equipment is worn out.

The depreciation rate can also be determined from

$$\frac{A}{P} = \frac{i}{1 - (1+i)^{-n}} = \frac{0.05}{1 - 1.05^{-15}} = 0.0963$$

In this formulation A represents the annuity that will both pay the earnings at i interest rate and repay the original investment P during the life of n years. In this case, since i is 5 per cent, the rate of repayment of original investment or the depreciation must be $= 0.0963 - 0.05 = 0.0463$, or 4.63 per cent, the same as determined by the sinking-fund formula.

Where income taxes apply, there is an increase in the cost of money use to realize a desired net rate of return. In most cases income taxes apply to the net income of an enterprise, i.e., that sum which is disbursed to dividends and surplus. The cost of money use as an equivalent interest rate to be used in economic evaluations should always be calculated according to the form in which the income tax is applied.

Example 29-10. A corporation has obtained 30 per cent of its capital from bond issues costing 3 per cent annually and 70 per cent from a stock issue on which it is desired to maintain a 5 per cent annual dividend. If annual income taxes of 40 per cent of net income must be paid, find the over-all cost of money use for the corporation as an equivalent interest rate.

Solution. For income-tax purposes, bond interest is an operating cost and not subject to the tax, but the net income of the business must be distributed to income tax, dividends, and surplus. Assuming that the surplus is negligible, then each 70 cents of \$1 of capital to maintain a net 5 per cent dividend must pay $0.7 \times 0.05 = 0.035$, or 3.5 cents annually. To pay the income tax in addition, however, it must earn $3.5/(1.0 - 0.4) = 5.83$ cents annually. The remaining 30 cents of each \$1 capital must earn $30 \times 0.03 = 0.9$ cent annually. Then the total cost of money use is $5.83 + 0.9 = 6.73$ cents per \$1 of capital, or an equivalent interest rate of 6.73 per cent.

In the practical field of power generation the total fixed-charge rate and its components as applied to power plants vary about as shown in the following tabulation.

	Per cent
Interest	4–8
Taxes	2–5
Insurance	0.05–0.3
Depreciation	1–5
Total	7–18

ANNUAL OPERATING COSTS

29-13. Costs. The operation of a power plant incurs expenses that vary with the extent of operation or magnitude of output. However, some of these costs remain constant as long as the plant is in active operation or is held in readiness to produce energy. These expenses are generally classified under the following headings: (1) fuel, (2) operating labor, (3) maintenance labor and materials, (4) supplies, (5) supervision, and (6) operating taxes.

29-14. Fuel. The largest item of expense in the operation of a thermal plant is the original raw energy in the form of fuel. This may be in the form of coal, oil, natural gas, wood scrap, or by-products. Where several of these are available, it will require careful investigation and analysis to determine which is the most advantageous. The fuel cost varies with the amount of energy produced, the efficiency of the plant, and the unit price of fuel.

29-15. Operating Labor. Supervision of equipment in a power plant is needed since no thermal plant can be made fully automatic. Although some hydraulic and gas-turbine plants are automatic in operation, they still require periodic inspection; operating labor is seldom completely eliminated as an item of expense. Great strides have been made in reducing the number of men required to operate a plant. This has been caused by two principal factors: (1) the increasing application of automatic equipment and (2) the increasing capacity of the various equipment units.

In a steam plant, labor is required for unloading and storing fuel and disposing of refuse. Men are needed for boiler operation, usually a crew to attend the combustion phase and another to attend the water being supplied the boiler. The operation of prime movers requires manual attention to start, stop, and watch over them during generation. Electrical switching and loading of the generators require another crew.

In the plant with only a few units of equipment the labor cost is likely to remain constant for all ranges of energy production; a plant containing a greater number of units requires somewhat less labor when operating at reduced outputs for extended periods of time. A hydraulic plant or internal-combustion (i-c) engine plant of a capacity equal to a steam plant will require less labor by reason of the smaller number of units of equipment that require attention.

29-16. Maintenance Labor and Materials. Any well-managed plant will generally follow a plan of preventive maintenance: inspecting, cleaning, and overhauling apparatus on a regular schedule to forestall the possibility of breakdown during service. This item of expense is made up of two factors: materials used in making a repair and labor required to effect it. Whenever a major repair or part replacement is required, it is a matter of arbitrary managerial policy whether it is charged to operating repairs or directly to the depreciation account as a replacement of original capital.

For purposes of estimating the maintenance of a proposed new plant, a review of the history of a similar plant in an analogous situation is used whenever possible. When there is no information on this score, the best that can be done is to assume an arbitrary percentage of fuel costs or other operating costs that vary with extent of operation.

29-17. Supplies. Supplies usually cover such items as water for make-up and general use, lubricating oils, water-treatment chemicals, tools, and wiping cloths. In general, any items that are not included in the categories of fuel or maintenance are charged to the supplies account.

29-18. Supervision. Supervision usually includes the salary of a station superintendent, chief engineer, chemist, and efficiency engineer. In some classifications this item also covers any materials that these men require.

29-19. Operating Taxes. Operating taxes include those which depend upon the magnitude of output rather than upon the size of the investment. These are taxes such as income, social security, and unemployment. Depending upon general economic conditions, the operating tax rate applied to operating costs may be three to four times as much as the tax rate applied to capital costs.

29-20. Total Operating Expense. Operating expenses in general cannot be expressed as a percentage of the investment since they will vary widely with the type of plant, its geographical location, and the rate of output. The range of percentage analysis of the individual items based upon the total operating cost for large plants varies as in the following tabulation.

	Per cent
Fuel	75–85
Labor	5–15
Maintenance labor and materials	5–20
Supplies	1–5
Supervision	1–2
Operating taxes	0–10

The range of analysis indicates that the fuel cost is the controlling factor in determining the magnitude of the total operating cost.

METHODS OF ECONOMIC SELECTION

29-21. General Factors. When the total energy needed per year and the maximum rate at which it will be demanded are known, the problem of selecting the means of supply arises. Theoretically this could be done with steam turbines, steam engines, i-c engines, gas turbines, or hydraulic turbines. Without going into economic cost considerations, it may be possible to eliminate some of these alternatives on the basis of other controlling factors such as nonavailability of water-power sites; unit capacities too small for size of demand as in reciprocating steam and i-c engines; inability to obtain certain fuels at economic prices in a particular locality; or lack of large water supply as required for a condensing turbine.

When narrowed down to one type of prime mover, the designer will be confronted with a wide variety of designs to choose from. The capital cost of the equipment usually increases with efficiency. In the case of a steam-turbine plant a choice may be made on the basis of a cycle such as a Rankine with steam-driven auxiliaries, or one to ten stages of regenerative feed heating, or a regenerative reheat. A choice also must be made on the basis of throttle pressure and temperature ranging anywhere from 200 psi, 500 F to 5,000 psi, 1200 F.

In selecting steam-generator equipment a choice must be made in regard to air preheaters and economizers. The choice of firing equipment will depend upon the fuels available and their relative costs.

29-22. Annual Costs. As equipment is added to improve station cycle efficiency, the total investment increases and with it the annual fixed costs, as illustrated in Fig. 29-1, the general relation being expressed by

$$C_f = RP$$

where C_f = annual fixed cost, dollars
R = fixed charge rate, as a decimal
P = capital investment, dollars

Whereas investment increases with efficiency, the benefit of such increase will be realized in lower fuel costs. Labor requirements would probably be alike in most cases with some increase as equipment is added.

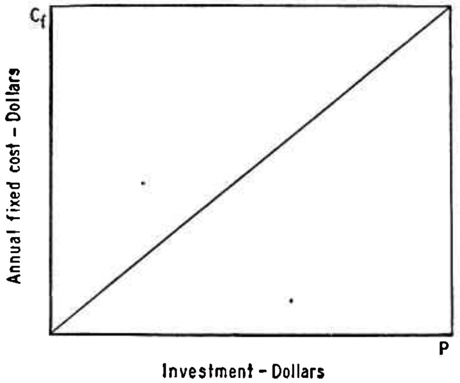

FIG. 29-1. Relation between annual fixed cost and investment for constant fixed-charge rate.

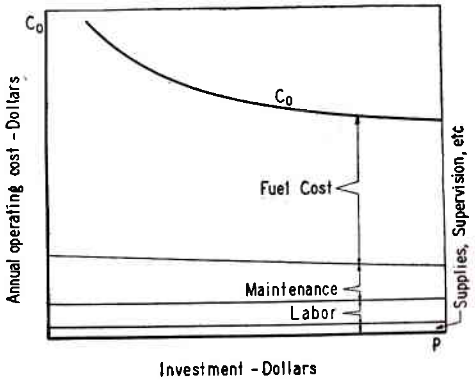

FIG. 29-2. General relation between annual operating cost and investments for alternate plants.

Operating supplies and supervision costs will probably remain constant, but maintenance and repair costs are generally a function of the amount of fuel burned. The general trends of these individual items are illustrated in Fig. 29-2. C_o is the total of all the annual operating costs.

To study the trend of the total annual cost C with investment, the two major component costs C_f and C_o are plotted on a common graph as in Fig. 29-3. Adding these two components at any given investment determines the characteristic of the total annual cost C, since $C = C_f + C_o$.

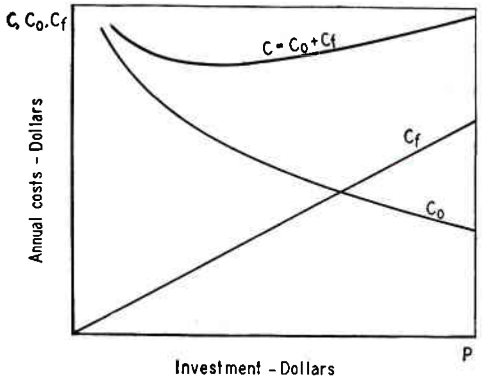

FIG. 29-3. General relation between components of total annual costs and investments for alternate plants.

At the range for the cheaper inefficient plants the operating cost is seen to be a major element in the total cost, whereas for the highly efficient plants the fixed cost becomes the major item. At some intermediate value of investment the total annual cost becomes a minimum.

The total cost becomes a minimum when the slope of the total-cost curve becomes zero, i.e., when

$$\frac{dC}{dP} = 0$$

and since

$$C = C_f + C_o$$

then

$$\frac{dC}{dP} = \frac{dC_f}{dP} + \frac{dC_o}{dP} = 0$$

hence

$$\frac{dC_o}{dP} = -\frac{dC_f}{dP}$$

or when the slopes of the fixed-cost curve and the operating-cost curve become equal numerically but opposite in sign. This does not happen necessarily where the two cost components are equal.

An enterprise can stay in business only if it sells its products at the lowest possible price. If the managers attempt to sell their products at a high profit or operate in an inefficient manner they soon find competitors entering the field and attempting to take their market away. Annual costs of a business and its income parallel each other; so aiming for minimum annual costs is the basic criterion for a successful business venture.

In power-generation activities minimum annual costs are achieved by a proper balance of the fixed and operating costs. The fundamental way to compare alternate schemes of power production is to compare their total annual costs.

Example 29-11. An industrial plant needs 50,000,000 kwhr of electrical energy a year with a maximum demand of 10,000 kw. These can be purchased from the local utility for $480,000 annually. As an alternate scheme the industry considers installing a 10,000-kw steam-turbine plant. The three plants of the following tabulation have been proposed:

	Plan *A*	Plan *B*	Plan *C*
Throttle steam conditions:			
Pressure, psig	200	400	900
Temperature, °F	550	700	850
Station steam rate (full load), lb per kwhr	11.8	10.5	8.5
Average station heat rate, Btu per kwhr	16,000	13,500	12,000
Unit installation costs:			
Steam generator and auxiliaries, dollars per lb-hr capacity	4.08	4.73	6.45
Steam turbine and auxiliaries, dollars per kw capacity	38.00	42.00	51.00
Electrical equipment	$200,000	$200,000	$200,000
Structures and miscellaneous	$200,000	$200,000	$200,000
Plant operators per shift	4	4	4

The station will run 24 hr a day with operators working 8 hr per day and 5 days a week. The average annual salary for operators is $5,200. Repair costs are estimated to be 90 cents per ton of coal burned for all schemes. From past experience the designers have found that ultimate installed costs exceed original estimates by 20 per cent; so this allowance is made as a contingency item. General operating supplies are estimated at $10,000 annually for all plans. Money earns at least 6 per cent in this business, which is expected to continue indefinitely, but the life of the plant is to be taken as 15 years.

Taxes on real estate and property amount to 4 per cent, and the various operating taxes add to 1 per cent of annual operating costs. Annual insurance premiums equal 0.2 per cent of all equipment costs. Fuel will be coal at $6.50 a short ton having a high heat value (HHV) of 14,200 Btu per lb.

Determine which scheme is the economic one, assuming all have equal reliability.

Solution

Boiler capacity = (station steam rate) × (station capacity)

Plan *A* = 11.8 × 10,000 = 118,000, say 120,000 lb per hr

Plan *B* = 10.5 × 10,000 = 105,000, say 110,000 lb per hr

Plan *C* = 8.5 × 10,000 = 85,000, say 90,000 lb per hr

Installed boiler cost = (boiler capacity) × (unit capacity cost)

Plan *A* = 120,000 × 4.08 = $490,000

Plan *B* = 110,000 × 4.73 = $520,000

Plan *C* = 90,000 × 6.45 = $580,000

Investment	Plan *A*	Plan *B*	Plan *C*
Boiler equipment	$ 490,000	$ 520,000	$ 580,000
Turbine equipment	380,000	420,000	510,000
Electrical equipment	200,000	200,000	200,000
Structures and miscellaneous	200,000	200,000	200,000
Total estimated items	$1,270,000	$1,340,000	$1,490,000
Contingencies at 20 per cent	250,000	270,000	300,000
Total investment	$1,520,000	$1,610,000	$1,790,000

$$\text{Depreciation rate} = \frac{A}{S} = \frac{i}{(1+i)^n - 1} = \frac{0.06}{1.06^{15} - 1} = 0.0431$$

$$\text{Fixed charge rate} = \underset{\overset{\downarrow}{\text{intere}}}{0.06} + \underset{\overset{\downarrow}{\text{deprec}}}{0.0431} + \underset{\overset{\downarrow}{\text{Taxes}}}{0.04} + \underset{\overset{\downarrow}{\text{insure}}}{0.002} = 14.51\%$$

Fuel cost:

$$\text{Unit fuel cost} = \frac{6.50 \times 10^6}{2{,}000 \times 14{,}200} = 22.9 \text{ cents per million Btu}$$

Annual total fuel cost:

$$\text{Plan } A = \frac{0.229}{10^6} \times 50{,}000{,}000 \times 16{,}000 = \$183{,}000$$

$$\text{Plan } B = \frac{0.229}{10^6} \times 50{,}000{,}000 \times 13{,}500 = \$155{,}000$$

$$\text{Plan } C = \frac{0.229}{10^6} \times 50{,}000{,}000 \times 12{,}000 = \$138{,}000$$

Operating labor cost:

There are 21 shifts per week, and any one man works 5 shifts per week. Then each position requires $^{21}\!/_5 = 4.2$ men to keep the post continuously manned. Assuming that operators are interchangeable in their posts, all plans will need 4 × 4.2 = 16.8, or 17 men.

Annual operating labor cost = 17 × 5,200 = $88,000

Maintenance costs annually:

$$\text{Plan } A = \frac{183{,}000}{6.50} \times 0.9 = \$25{,}000$$

$$\text{Plan } B = \frac{155{,}000}{6.50} \times 0.9 = \$21{,}000$$

$$\text{Plan } C = \frac{138{,}000}{6.50} \times 0.9 = \$19{,}000$$

Annual operating costs	Plan *A*	Plan *B*	Plan *C*
Fuel	$183,000	$155,000	$138,000
Operating labor	88,000	88,000	88,000
Maintenance	25,000	21,000	19,000
Supplies	10,000	10,000	10,000
Subtotal	$306,000	$274,000	$255,000
Operating taxes at 1 per cent	3,000	3,000	3,000
Total annual operating costs	$309,000	$277,000	$258,000

Total annual costs	Plan *A*	Plan *B*	Plan *C*	Utility service
Investment charges at 14.5 per cent	$221,000	$233,000	$260,000	
Operating costs	309,000	277,000	258,000	
Total annual costs	$530,000	$510,000	$518,000	$480,000

For the given conditions of fixed-charge rate and fuel costs, plan *B* incurs the least annual costs of the three station plans. However, since the utility supply can be obtained for less than any of the private-plant supplies, it would be the logical choice. In certain circumstances, even if the utility service cost $520,000, it might be selected in preference to the private-plant plan *B*. An industrial organization can usually make its greatest return by investing its capital funds directly in its main production facilities. Any money tied up in auxiliary equipment, such as power-generation apparatus, diverts capital funds from a possible expansion of the principal producing plant. Thus, in some instances the conclusions drawn strictly from a financial study may be modified by other practical considerations.

If no utility service were available, plan *B* would undoubtedly be chosen for installation.

29-23. Influence of Fixed-charge Rate. The magnitude of the fixed-charge rate is controlled principally by the interest rate and the life span assumed. Economic comparisons and conclusions are necessarily influenced by the magnitude of these elements. In the foregoing example assume that the fixed-charge rate is 7 per cent instead of 14.51 per cent, and observe the effect on the results.

Total annual costs	Plan *A*	Plan *B*	Plan *C*
Investment charges at 7 per cent	$106,000	$113,000	$125,000
Operating costs	309,000	277,000	258,000
Total annual cost	$415,000	$390,000	$383,000

The reduction in fixed-charge rate would be brought about by a smaller cost of money use, or a longer life span, or both. In this case plan *C* is the optimum installation. Comparing these results with those of the previous example an important principle can be deduced: as the fixed-charge rate decreases, the plans involving higher capital investment are favored as optimum installations. In other words, as the cost of money use decreases or the life span of the investment increases, more money may be invested to secure greater over-all economy. This principle holds only when the annual operating costs of each alternative remain unchanged.

29-24. Influence of Annual Operating Costs. For all practical considerations in thermal stations the value of the fuel controls the general magnitude of the annual operating costs. Thus its purchase price will have a controlling effect on the economic conclusions to be drawn from financial comparisons. In Example (29-11) assume that the coal cost is doubled to $13 per short ton, and observe the effect on the comparison.

Total annual costs	Plan *A*	Plan *B*	Plan *C*
Investment charges at 14.5 per cent	$221,000	$233,000	$260,000
Operating costs with $13 coal	492,000	432,000	396,000
Total annual costs	$713,000	$665,000	$656,000

For this condition the plan requiring the highest investment again becomes the optimum of the three. From this a corollary principle may be deduced: as the cost of fuel increases, plans involving higher capital investment are favored as optimum installations. In other words, as the cost of fuel increases, more money may be invested to gain increased efficiency. This holds only when the fixed-charge rate is common to all schemes and remains constant.

29-25. Rate of Return. Though the computation of total annual costs must be made in most cases and a comparison of total annual costs is the fundamental method of making economic comparisons, other methods have been developed of making these comparisons. Though expressed differently, the same answer must be obtained for all methods. A popular method treats the rate of return or interest rate to be realized from the various investments as an unknown. This method is best shown by an example.

Example 29-12. Assume the same data as in Example 29-11 except that the utility service costs $530,000 annually. Then all the items of annual costs are computed except depreciation and interest. The difference between the utility-service cost of $530,000 and the annual costs that are computed represents a saving available for the annual interest on the investment and its amortization. These differences will be representative of the annuity A and the investment will be the principal P. Then i will be solved for by trial and error from

$$\frac{A}{P} = \frac{i}{1 - (1 + i)^{-n}}$$

Solution

	Plan A	Plan B	Plan C	Utility service
Annual operating costs	$309,000	$277,000	$258,000	
Annual taxes on capital	61,000	64,000	72,000	
Annual insurance costs	3,000	3,000	4,000	
Annual costs excluding interest and depreciation	$373,000	$344,000	$334,000	$530,000
Difference available for annuity A	$157,000	$186,000	$196,000	Base
Investment P	$1,520,000	$1,610,000	$1,790,000	
Ratio A/P	0.1033	0.1155	0.1094	
Rate of return i, per cent	6.4	7.8	7.0	

All plans yield a greater rate of return than the minimum interest rate of 6 per cent. First this means that it will be desirable to build one of these plants rather than purchase the energy at a cost of $530,000 a year; second, although plan B yields the maximum rate of return on the investment, it seems that a very attractive rate of return can be earned on the higher investment by plan C. The limit of investment is determined by that value where the last additional dollar invested earns at least the minimum rate of return desired, in this case 6 per cent. This method investigates the *rate of return on increment investment.* It consists in applying the same principles outlined in the preceding discussion but finding the return earned on the differences of investment for the various plans.

	Difference of plans	
	B over A	C over B
Saving in annual costs, excluding interest and depreciation, ΔA	$29,000	$ 10,000
Additional capital investment, ΔP	$90,000	$180,000
Ratio $\Delta A/\Delta P$	0.3222	0.0556
Rate of return on increment investment,[a] per cent	31.7	Negative

[a] By trial solution for i from

$$\frac{\Delta A}{\Delta P} = \frac{A}{P} = \frac{i}{1 - (1 + i)^{-n}}$$

or interpolating in Appendix Table 16.

Although plan A showed a satisfactory return of 6.4 per cent, the additional investment of $90,000 for plan B earns the phenomenal rate of 31.7 per cent, which raises the over-all return of plan B to 7.8 per cent. The additional investment of $180,000 for plan C over plan B does not earn even enough to amortize it, as indicated by the negative interest rate. This deficient return reduces the over-all return of plan C to 7.0 per cent Hence plan B is the optimum scheme for the given conditions. The same conclusion was reached by the total-annual-cost method.

The importance of thoroughly investigating all possibilities in economic engineering is illustrated in the following. If Plan B had been overlooked, the return of the additional investment for C over A would have been as shown in the following tabulation.

Saving in annual costs, excluding interest and depreciation, ΔA	$ 39,000
Additional capital investment ΔP	$270,000
Ratio $\Delta A/\Delta P$	0.1445
Rate of return on increment investment, per cent	11.7

29-26. Return Available for Fixed Charges. This method is frequently applied since it eliminates the involved method of cut-and-try solution for the interest rate needed in the rate-of-return method. The fixed-charge rate is readily calculated from the desired rate of return and life of the project including, of course, the tax and insurance-premium rates. From the proposals studied the savings as a percentage of the corresponding investment that are available to meet fixed charges can be determined. It will again be necessary to study the rates for the increment investment to arrive at the optimum scheme. This is illustrated in the following, using the same example as in the foregoing section.

Solution

	Plan A	Plan B	Plan C	Utility service
Annual operating costs	$ 309,000	$ 277,000	$ 258,000	$530,000
Savings available for investment charges	$ 221,000	$ 253,000	$ 272,000	Base
Investment	$1,520,000	$1,610,000	$1,790,000	0
Ratio of savings to investment,[a] per cent	14.6	15.7	15.2	
Increment savings		$32,000	$ 19,000	
Increment investment		$90,000	$180,000	
Ratio of increment savings to increment investment, per cent[a]		35.6	10.6	

[a] Rates comparable to fixed-charge rate = 14.51 per cent.

Here again all the plant schemes show a greater availability of savings for investment charges than is basically required. However, the rate for the increment investment of C over B shows this added expenditure to be unprofitable. Hence plan B is the optimum one. This is the conclusion drawn from the previous methods of comparison.

29-27. Justifiable Investment. This method is a variation of the return available for fixed charges. Here the savings available for investment charges are divided by the fixed-charge rate to arrive at a justifiable maximum investment. The justifiable (maximum) investment is compared with the actual investment required for the given plan; if the actual investment is equal to or less than the justifiable investment, then the plan may prove satisfactory. Again the limit of investment must be determined. This is termed the *justifiable added investment* and is found by dividing the increment saving between alternate plans by the fixed-charge rate. If the actual increment investment is equal to or less than the justifiable added investment, then the higher-cost plan is acceptable; otherwise the added investment will not earn the required return. This method is illustrated by the example used in the preceding section.

	Plan A	Plan B	Plan C	Utility service
Annual operating costs	$ 309,000	$ 277,000	$ 258,000	$530,000
Savings available for investment charges	$ 220,000	$ 253,000	$ 272,000	Base
Fixed-charge rate, per cent	14.5	14.5	14.5	
Justifiable investment	$1,525,000	$1,745,000	$1,877,000	
Actual investment needed	$1,520,000	$1,610,000	$1,790,000	
Increment savings		$ 32,000	$ 19,000	
Fixed-charge rate, per cent		14.5	14.5	
Justifiable added investment		$221,000	$131,000	
Actual increment investment needed		$ 90,000	$180,000	

For schemes A, B, and C the actual investments needed are less than the justifiable (maximum) investments and hence appear to be satisfactory. The difference between these two items, however, varies for each plan: for A it is $5,000, for B $135,000, and for C $87,000. The optimum plan is revealed by comparing the justifiable *added* investments and the actual *increment* investments. For B over A we can spend as much as $221,000 but actually only need $90,000. For C over B, however, we can justify only $131,000 extra but actually need $180,000. Hence plan C is not relatively economical and plan B is the optimum scheme. This conclusion was also reached by the previous methods of comparison.

29-28. Present Worth. This method and the capitalized-cost method described in the succeeding section may be regarded as index methods of comparison. The preceding methods are all rational in their approach in that the figures are directly comparable with an actual expended sum of money. In the present-worth method all the annual costs expended during the life of the project or equipment are reduced to that sum which invested now at a given interest rate would yield an annuity equal to the series of annual costs or expenditures. The usual interest rate is the one that measures the cost of money use for the enterprise. Letting t = the tax rate on capital equipment and j = the insurance rate, and since

$$
\begin{aligned}
C &= C_f + C_o = RP + C_o \\
&= \left[i + \frac{i}{(1+i)^n - 1} + t + j \right] P + C_o
\end{aligned}
$$

then the present worth of the series of annual expenditures C for n years is

$$
\begin{aligned}
W_p &= \frac{\left[i + \dfrac{i}{(1+i)^n - 1} + t + j \right] P + C_o}{i/[1 - (1+i)^{-n}]} \\
&= P + \frac{(t+j)P + C_o}{i/[1 - (1+i)^{-n}]}
\end{aligned}
$$

It follows that the scheme with the lowest present worth involves a minimum over-all expenditure during life and therefore would be the desirable or optimum plan. This method is illustrated in the following, again using the data of the previous example.

Solution

$$
\frac{i}{1 - (1+i)^{-n}} = \frac{0.06}{1 - 1.06^{-15}} = 0.103
$$

	Plan A	Plan B	Plan C	Utility service
Annual operating costs	$ 309,000	$ 277,000	$ 258,000	$ 530,000
Annual taxes on capital	61,000	64,000	72,000	
Annual insurance costs	3,000	3,000	4,000	
Annual costs excluding interest and depreciation	$ 373,000	$ 344,000	$ 334,000	$ 530,000
Present-worth factor	0.103	0.103	0.103	0.103
Present worth of costs excluding interest and depreciation	$3,620,000	$3,340,000	$3,240,000	$5,150,000
Investment P	1,520,000	1,610,000	1,790,000	0
Total present worth W_p	$5,140,000	$4,950,000	$5,030,000	$5,150,000

Plan B, having the minimum present worth, entails the least total expenditures over the life of 15 years and is therefore the optimum one. This is the answer obtained by the previous methods.

29-29. Capitalized Cost. This method of comparison is very similar to the present-worth method. In the present-worth method, the assumption of a lump sum of money covers only the actual schedule of total annual costs to be met during the life span assumed. The fundamental criterion in the capitalized-cost method is that the enterprise will continue forever. Hence the annual costs will continue forever, and the initial investment P will have to be made at the end of every n years forever. The capitalized cost is that sum of money which would supply forever the infinite series of annual costs. Such a sum would be that one on which the annual earnings or interest would just cover the total annual costs of the enterprise as defined above. If W_c is the capitalized cost, then

$$iW_c = \left[i + \frac{i}{(1+i)^n - 1} + t + j\right]P + C_o$$

and

$$W_c = P + \frac{\left[\dfrac{i}{(1+i)^n - 1} + t + j\right]P + C_o}{i}$$

As in the present-worth method, it follows that the plan with the lowest capitalized cost is the optimum one for the given circumstances. This is illustrated in the following, using the same data as in the previous demonstrations.

Solution

	Plan A	Plan B	Plan C	Utility services
Annual operating costs	\$ 309,000	\$ 277,000	\$ 258,000	
Annual depreciation costs	66,000	70,000	77,000	
Annual taxes on capital	61,000	64,000	72,000	
Annual insurance costs	3,000	3,000	4,000	
Annual costs excluding interest	\$ 439,000	\$ 414,000	\$ 411,000	\$ 530,000
Interest rate i	0.06	0.06	0.06	0.06
Capitalized costs of annual costs excluding interest	\$7,310,000	\$6,900,000	\$6,850,000	\$8,833,000
Investment P	1,520,000	1,610,000	1,790,000	0
Total capitalized cost W_c	\$8,837,000	\$8,517,000	\$8,640,000	\$8,833,000

Plan B, having the minimum capitalized cost, entails the smallest annual costs and is therefore the optimum plan. Again this is the answer obtained by all the previous methods.

29-30. Summary. For easy use in problems given here and following chapters we collect here all the comparison methods just discussed:

1. *Total annual cost.*

$$\begin{aligned} C &= C_f + C_o \\ &= RP + C_o \\ &= \left[i + \frac{i}{(1+i)^n - 1} + t + j\right]P + C_o \end{aligned}$$

where C_o is made up of elements $C_{oa} + C_{ob} + C_{oc} + \cdots + C_{oz}$.

The plan with the lowest annual cost C is the optimum.

NOTE: In considering alternate plans, if any of the individual operating costs $C_{oa,ob,oc,\ldots,oz}$ are identical for all plans, it is not necessary to include them in the total cost C for all the plans, since partial totals without these equal costs will also determine the minimum cost plan.

2. *Rate of return.* Solve for i by trial and error from

$$\frac{A}{P} = \frac{i}{1 - (1+i)^{-n}}$$

where $A = Q - [C_o + P(t + j)]$

Q = total annual cost of some other alternate plan not requiring a comparable initial investment

Plans with i equal to or greater than the minimum required i may be acceptable, provided that they pass the test of *rate of return on increment investment* summarized below.

NOTE: All costs must be included in this method.

3. *Rate of return on increment investment.* Solve for i by trial and error from

$$\frac{\Delta A}{\Delta P} = \frac{i}{1 - (1+i)^{-n}}$$

where $\Delta A = [C_o' + P'(t + j)] - [C_o'' + P''(t + j)]$

$\Delta P = P'' - P'$

The plan with the higher investment is acceptable if i is equal to or greater than the minimum required value of i; otherwise the plan with the lower investment is the optimum *of the two*. Where more than two plans are studied, the compairsons must be made between plans of successively higher investments.

NOTE: C_o' and C_o'' need not include cost elements that are equal for both plans.

4. *Rate of return available for fixed charges.*

$$R = \frac{Q - C_o}{P}$$

The plans with R equal to or greater than the required R may be acceptable, provided that they pass the test of *rate of return available for fixed charges on increment investment* summarized below.

NOTE: All costs must be included in this method.

5. *Rate of return available for fixed charges on increment investment.*

$$R = \frac{C_o' - C_o''}{P'' - P'}$$

The plan with the higher investment is acceptable if R is equal to or greater than the minimum required value of R; otherwise the plan with the lower investment is the optimum *of the two*. Where more than two plans are studied, the comparisons must be made between plans of successively higher investment.

NOTE: C_o' and C_o'' need not include cost elements that are equal for both plans.

6. *Justifiable investment.*

$$P = \frac{Q - C_o}{R}$$

The plan with justifiable investment P equal to or greater than the actual required investment may be acceptable if it passes the test of *justifiable added investment* summarized below.

NOTE: All costs must be included in this method.

7. *Justifiable added investment.*

$$\Delta P = \frac{C_o' - C_o''}{R}$$

The plan with the higher investment is acceptable if the justifiable added investment ΔP is equal to or greater than $P'' - P'$ actually required; otherwise the plan with the lower investment is the optimum *of the two*. Where more than two plans are studied, the comparisons must be made between plans of successively higher investments.

NOTE: C_o' and C_o'' need not include cost elements that are equal for both plans.

8. *Present worth.*

$$W_p = P + \frac{P(t + j) + C_o}{i/[1 - (1 + i)^{-n}]}$$

The plan with the lowest present worth is the optimum.

NOTE: Equal costs of all plans need not be included in this method.

9. *Capitalized cost.*

$$W_c = P + \frac{\left[\dfrac{i}{(1 + i)^n - 1} + t + j\right] P + C_o}{i}$$

The plan with the lowest capitalized cost is the optimum.

NOTE: Equal costs of all plans need not be included in this method.

PROBLEMS

29-1. Find the sum of money that would be accumulated in a 10-year period by investing $500 at 4 per cent interest.

29-2. Find the compound amount of $200 invested at 3 per cent for 5 years.

29-3. Find the present worth of $1,800 twenty years from now if money can earn 5 per cent.

29-4. What payment may be accepted now in lieu of a payment 15 years hence of $400 if interest is at 2 per cent?

29-5. Find the present worth of $1,500 twelve years hence if 5 per cent interest is compounded quarterly.

29-6. What is the accumulation in 8 years of $200 invested at 2.5 per cent compounded monthly?

29-7. What is the present worth of $13,000 paid 100 years from now if the interest rate is 10 per cent?

29-8. What interest rate was earned on an original loan of $1,000 if $1,708 was repaid 10 years later?

29-9. If $100 is invested at 7 per cent, how many years will be required to triple its original value?

29-10. How much must be invested at the end of each year to accumulate $10,000 at the end of 20 years if interest is at 6 per cent?

29-11. What annual payment is justified in order to avoid spending $5,000 fifteen years from now if money earns 4.5 per cent?

29-12. What is the accumulation of investing $50 monthly for 14 years if the nominal interest rate is 8 per cent?

29-13. What annual year-end payment will repay a present loan of $7,000 in 20 years if interest is 5.5 per cent?

29-14. Find the present worth of $100 paid quarterly for a period of 15 years if money earns 4 per cent.

29-15. What annual payment might be secured for an infinite period from a sum of $1,000 invested at 8 per cent interest?

29-16. In order to secure a perpetual income of $1,000 annually, what investment must be made at 6 per cent interest?

29-17. A corporation has sold an issue of bonds maturing in 15 years and having a face value of $3,500,000 to an investment syndicate for $3,180,000. The bonds bear an interest rate of 3.5 per cent payable semiannually. Legal fees and other preparatory expenses for floating the loan have cost the corporation $60,000. If the clerical and bookkeeping work associated with paying the interest involves an expense of $5,000 per interest period, what is the cost of using the money realized from these bonds to the corporation expressed as an interest rate?

29-18. If the corporation in Prob. 29-17 has raised $4,500,000 by the sale of stock on which it expects to pay 5 per cent quarterly, what is the average cost of money use, or interest rate, that it should use in calculating its annual costs?

29-19. What is the amortization or depreciation rate for an enterprise that is earning 6 per cent on its invested funds and estimates a life span of 20 years? Salvage value of the equipment will pay for its disposal. Do this by two methods.

29-20. An electric utility is required by law to allocate from its annual earnings a sum equivalent to 1.5 per cent of its plant investment for depreciation. If the cost of money use to this utility is 6 per cent, what is the implied average life of the plant on the basis of sinking-fund depreciation?

29-21. If the plant of the utility in Prob. 29-20 is assessed at 80 per cent of first cost and the tax rate is 5 per cent of assessed valuation, and if equipment insurance is 0.2 per cent of the first cost, what is the plant fixed-charge rate?

29-22. Six alternate plans of supplying energy for a given circumstance are investigated and found to require the investments and annual operating costs given in Table P-1.

TABLE P-1

	Plan *A*	Plan *B*	Plan *C*	Plan *D*	Plan *E*	Plan *F*
Investment	$50,000	$100,000	$150,000	$200,000	$250,000	$300,000
Annual operating cost	36,000	27,000	20,000	15,000	12,000	10,000

a. Find the optimum plan for a fixed charge rate of 10 per cent.

b. Find the optimum plan for a fixed charge rate of 5 per cent.

c. Find the optimum plan for a fixed charge rate of 15 per cent.

d. Compare the total annual costs for the optimum plans of (*a*), (*b*), and (*c*), and discuss.

e. Assuming that the operating costs are 20 per cent greater than given, find the optimum plan at a fixed charge rate of 10 per cent.

f. Assuming that the operating costs are 20 per cent less than given, find the optimum plan at a fixed charge rate of 10 per cent.

g. Compare the total annual costs for the optimum plans of (*a*), (*e*), and (*f*), and discuss.

29-23. A study is being made to determine the economic diesel plant to serve a given energy demand. Three plans are under consideration, the life of each of which is assumed to be 20 years. The cost of money use is 6 per cent; taxes and insurance cost a total of 5 per cent of investment. Other costs are as in Table P-2.

TABLE P-2

	Plan *A*	Plan *B*	Plan *C*
Investment	$220,000	$235,000	$245,000
Annual fuel cost	$ 31,800	$ 21,000	$ 20,400
Annual lubrication cost	$ 1,000	$ 2,000	$ 2,000
Annual labor cost	$ 10,000	$ 12,000	$ 12,000
Annual maintenance cost	$ 3,000	$ 5,000	$ 5,000

a. Determine the optimum plan on the basis of total annual costs.

b. Determine the optimum plan on the basis of rate of return on increment investment.

c. Determine the optimum plan on the basis of rate of return available for fixed charges on increment investment.

d. Determine the optimum plan on the basis of added justifiable investment.

e. Determine the optimum plan on the basis of present worth of each scheme.

f. Determine the optimum plan on the basis of capitalized costs.

NOTE: In this problem the plant must be built since no alternative service is available. Hence the methods of rate of return on total investment, rate of return available for fixed charges on total investment, and total justifiable investment cannot be applied.

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CHAPTER 30

LOAD CURVES

30-1. Defining Energy Requirements. In any problem of power-plant design the first step defines the conditions of energy supply that the plant must meet. The conditions may be classified under three headings: (1) maximum demand, (2) total energy requirements, and (3) distribution of energy demand.

30-2. Maximum Demand. The capacity of a plant depends upon the maximum power demand that will be made by the energy-consuming devices it feeds. Almost any plant will supply a variety of services with varying demands over a period of time. Such devices are machine-drive motors, lights, process and space heating, elevators, conveyers, and many others.

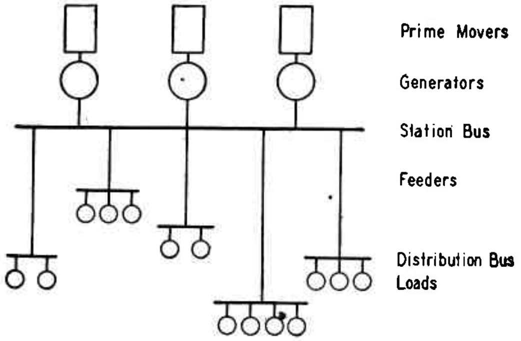

FIG. 30-1. Single-line diagram of an elementary power system.

The principal elements of a power system may be indicated as in Fig. 30-1. The individual energy-consuming devices or loads each connects to a distribution bus as a source of supply. All the distribution busses receive their energy from the station bus through feeders. The station bus is supplied by one or more generators driven by direct-coupled prime movers. Electrical energy in the central station is usually generated at high potential varying from 4 to 18 kv in most cases. Transmission of the energy over the feeders may take place at potentials as high as 400 kv. The longer the distance, the higher the line voltage employed. This tends to conserve on copper utilization, but this saving is offset by more elaborate towers and insulation requirements. At the distribution busses the potential is transformed to the voltage required by the consuming devices.

Each device has a maximum capacity for absorption of power. For the usual case each device of a group operated by an individual consumer can function independently. If all the devices ran to their fullest extent simultaneously, the *maximum demand* of the consumer on the energy system would equal his *connected load.* However, experience demonstrates that the actual maximum demand of a consumer will be less than his connected load since all the devices never run at full load at the same time. The relationship of maximum demand and connected load is measured by

$$\text{Demand factor} = \frac{\text{maximum demand}}{\text{connected load}}$$

The demand factor depends upon the nature and activities of the consumer and to some extent upon the location of the power system. Various studies indicate that demand factors may vary from about 25 per cent for hotels to 90 per cent for refrigeration plants. Each device will reach its own maximum demand at some time during its operation, but the demand factor measures the extent to which it contributes toward the maximum demand of a group of devices of which it is a part.

Experience shows that the maximum demands of individual consumers do not occur simultaneously but are spread out over a period of time. This holds for consumers whose activities and energy requirements are very similar. The time distribution of maximum demands for similar types of consumers is measured by the diversity factor, where

$$\text{Group diversity factor} = \frac{\text{sum of individual maximum demands}}{\text{actual maximum demand of group}}$$

The group diversity factor is always greater than unity. Diversity factors for residential consumers are usually highest at about 5.0, whereas large industrial consumers may have values as low as 1.3. Since diversity exists between individual maximum demands, the proportion contributed to the *system* maximum demand by each consumer is less than his maximum demand.

The peak demand of a system is made up of the individual demands of the devices that happen to be functioning at the time of the peak. At the time of the system peak demand the demand of a particular group of similar consumers is seldom at the maximum value that it may reach at some other time of the year. This diversity is measured by

$$\text{Peak diversity factor} = \frac{\text{maximum demand of consumer group}}{\text{demand of consumer group at time of system peak demand}}$$

The method of determining the maximum demand to be expected on a system when the foregoing data are available is indicated below.

Let

$C_1, C_{1'}, C_{1''}, \ldots, C_{1^n}$ = individual connected loads of group 1
$C_2, C_{2'}, C_{2''}, \ldots, C_{2^n}$ = individual connected loads of group 2
d_1 = demand factor of group 1
d_2 = demand factor of group 2
D_1 = group diversity factor among consumers in group 1
D_2 = group diversity factor among consumers in group 2
M_1 = maximum demand of group 1
M_2 = maximum demand of group 2
r_1 = peak diversity factor for group 1
r_2 = peak diversity factor for group 2
L_m = system maximum demand
$L_1, L_2, L_3, \ldots, L_n$ = demands of each type of consumer at time of system maximum demand

Then

$$M_1 = \frac{\Sigma C_1 \times d_1}{D_1}$$

$$M_2 = \frac{\Sigma C_2 \times d_2}{D_2}$$

$$L_1 = \frac{M_1}{r_1}$$

$$L_2 = \frac{M_2}{r_2}$$

$$L_m = L_1 + L_2 + L_3 + \cdots + L_n$$

The demand and diversity factors will be chosen from similar situations in existing systems when such data are available. When this is not the case, each power requirement will have to be studied in detail and the hourly variation in demand estimated throughout the operating period of each device for all days during which the peak is likely to occur; when the energy requirements at each hour of the day for all the individual requirements are summed up, the variation in hourly total demand can be determined. By studying typical days for various parts of the year the maximum demand can be found.

The success of the central-station system of energy supply stems from the diversity of the demands of the various components of the total connected load. The capacity required to supply such a system is much less than would have to be provided if each load component had its own plant. In addition the central system uses larger-capacity units of generating equipment that are more efficient and less costly than a multiplicity of small units.

Example 30-1. A new housing development is to be added to the lines of a public utility. There are 1,000 apartments, each having a connected load of 4 kw; also stores and services are included of the characteristics shown in the following tabulation:

Store or service	Connected kw	Demand factor, per cent
1 laundry	20	68
2 churches	10 each	56
1 restaurant	60	52
1 bookstore	5	66
1 dry-goods store	7	76
2 drugstores	10 each	79
2 grocery stores	5 each	73
1 shoe store	2	67
1 clothing store	4	53
1 theater	100	49

The demand factor of the apartments is 45 per cent. The group diversity factor of the residential load for this system is 3.5, and the peak diversity factor is 1.4. The commercial-load group diversity factor is 1.5, and the peak diversity factor is 1.1.

Find the increase in peak demand on the total system delivery from the station bus resulting from addition of this development on the distribution system. Assume line losses at 5 per cent of delivered energy.

Solution

a. Maximum demand per apartment $= 4 \times 0.45 = 1.8$ kw

b. Maximum demand of 1,000 apartments $= \dfrac{1.8 \times 1{,}000}{3.5} = 514$ kw

c. Demand of 1,000 apartments at time of system peak $= \dfrac{514}{1.4} = 367$ kw

d. Commercial loads (including churches) as in the following tabulation:

Service	Connected kw	Demand factor	Maximum demand, kw
Laundry	20	0.68	13.6
Churches	20	0.56	11.2
Restaurant	60	0.52	31.2
Bookstore	5	0.66	3.3
Dry-goods store	7	0.76	5.3
Drugstores	20	0.79	15.8
Grocery stores	10	0.73	7.3
Shoe store	2	0.67	1.3
Clothing store	4	0.53	2.1
Theater	100	0.49	49.0
Total of commercial maximum demands			140.1

e. Maximum demand of commercial group $= \dfrac{140.1}{1.5} = 94$ kw

f. Commercial demand at time of system peak $= \dfrac{94}{1.1} = 86$ kw *g.* Demand at point of consumption at time of system maximum demand = 367 + 86 = 453 kw

h. Total increase in maximum demand at station bus = 453 × 1.05 = 476 kw

Example 30-2. A small manufacturing plant making a new product requires electrical energy service. To determine the capacity needed, the operations of two critical days are estimated, a winter day and a hot summer day. Hourly energy requirements for each function are estimated and the totals found as indicated in the accompanying tabulation.

Solution

	A.M.												P.M.											
	1	2	3	4	5	6	7	8	9	10	11	12	1	2	3	4	5	6	7	8	9	10	11	12
Winter day, kw:																								
Lighting	1	1	1	1	1	3	4	5	5	5	5	5	5	5	5	7	15	15	5	5	3	1	1	1
Heating	2	2	2	2	3	7	10	10	10	9	8	7	7	7	7	8	7	6	6	6	6	1	1	1
Pumping	0	0	0	0	0	0	0	30	30	30	30	30	30	30	30	30	30	0	0	0	0	0	0	0
Stamping	0	0	0	0	0	0	0	20	20	40	40	0	20	40	40	40	0	0	0	0	0	0	0	0
Machining	0	0	0	0	0	0	0	5	5	5	5	0	5	5	5	5	0	0	0	0	0	0	0	0
Refrigeration	4	4	4	4	4	4	4	4	5	5	5	4	4	4	4	4	4	4	4	4	4	4	4	4
Conveyers	0	0	0	0	0	0	0	2	2	2	2	0	2	2	2	2	2	0	0	0	0	0	0	0
Shipping	1	1	1	1	1	1	1	1	1	1	1	1	7	7	7	7	7	7	7	7	5	2	1	1
Total demand, kw	8	8	8	8	9	15	19	77	78	97	96	47	80	100	100	103	65	32	22	22	18	8	7	7
Summer day, kw:																								
Lighting	1	1	1	1	1	3	4	5	5	5	5	5	5	5	5	5	5	5	5	5	3	1	1	1
Fans	0	0	0	0	0	0	0	0	2	2	2	4	5	7	7	7	2	2	2	2	0	0	0	0
Pumping	0	0	0	0	0	0	0	30	30	30	30	30	30	30	30	30	30	0	0	0	0	0	0	0
Stamping	0	0	0	0	0	0	0	20	20	40	40	0	20	40	40	40	0	0	0	0	0	0	0	0
Machining	0	0	0	0	0	0	0	5	5	5	5	0	5	5	5	5	0	0	0	0	0	0	0	0
Refrigeration	6	6	6	6	6	6	6	6	8	8	8	9	11	12	12	14	14	12	10	9	8	8	8	7
Conveyers	0	0	0	0	0	0	0	2	2	2	2	0	2	2	2	2	2	0	0	0	0	0	0	0
Shipping	1	1	1	1	1	1	1	1	1	1	1	1	7	7	7	7	7	7	7	7	5	2	1	1
Total demand, kw	8	8	8	8	8	10	11	69	73	93	93	49	85	108	108	110	60	26	24	23	16	11	10	9

On examining the hourly total demands it is found that at 4 P.M. on a summer day the peak of 110 kw occurs.

30-3. Variation in Demand.

The chronological variation in demand for energy on the source of supply is plotted graphically for study and easy comprehension. Such graphs for electrical energy demands (Fig. 30-2) are generally termed *load curves.* The curve for the industrial load typifies a factory operation on a one-shift basis. In the early morning hours the demand is generally for lighting and auxiliary drive for the heating boiler plant as well as some processes that require continuous energy supply such as refrigeration and electric furnaces. About 5 A.M. some of the factory machinery starts running, perhaps for warming prior to operation, or it may be that one department starts earlier in the day than the remainder of the organization to synchronize the work properly. By 8 A.M., in this particular case, the entire plant is running, and energy

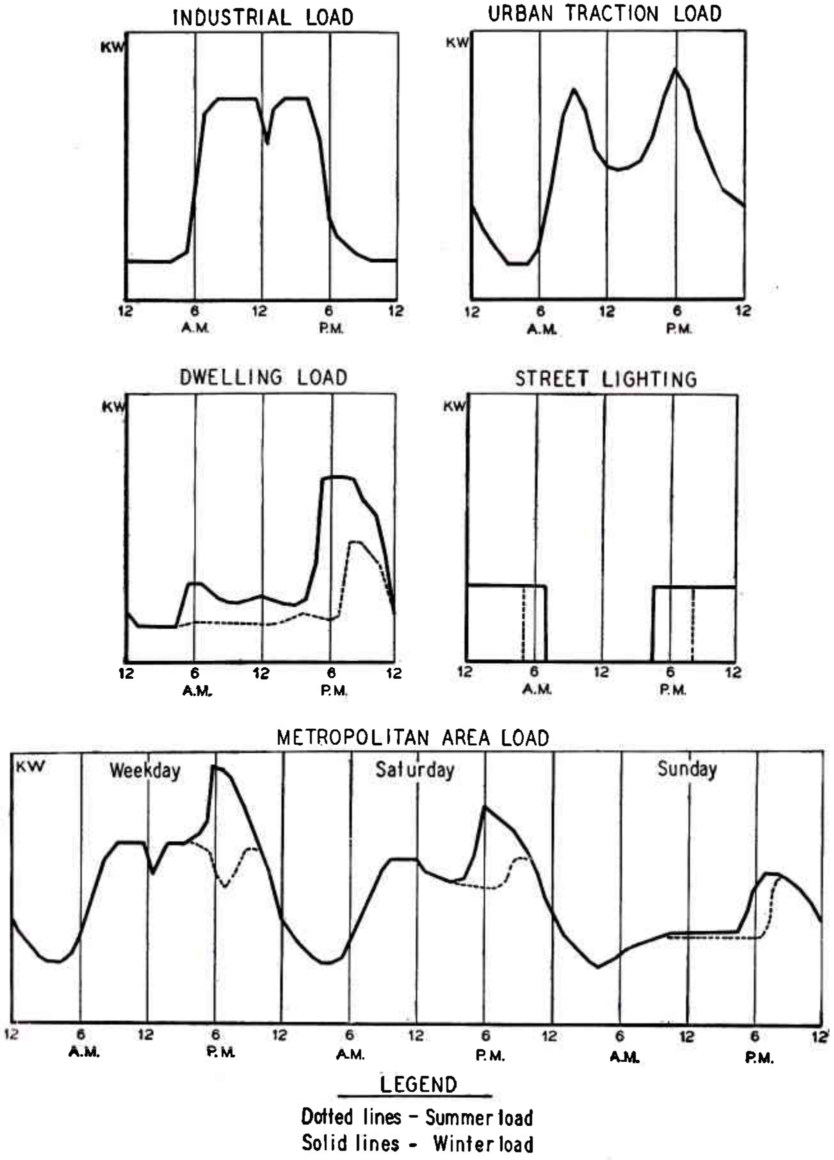

FIG. 30-2. Typical hourly variation in energy demand for different types of electric loads.

demand remains substantially constant until shortly before noon. Load falls off as some of the machines shut down during the lunch period. The extent of the noon-hour dip depends upon the individual factory. By 2 P.M. the load attains about the same level as during the morning. Shortly before 4 P.M. the load starts to drop as the day's work ends in the various departments. At 6 P.M. most of the machines are shut down, and the load gradually tapers off until about 9 or 10 P.M., when the minimum demand is reached and continues until the start of the next working day.

The curve for the urban traction load is typical of railroads operating within the larger cities. From midnight till 3 A.M. the demand tapers off as service reaches its minimum level, which continues until about 5 A.M. in this particular instance. This load is largely lighting, with power at a minimum since only a few trains run. As the early factory workers start for their work, service rapidly increases and the consequent load continuously rises as these travelers are augmented and followed by office workers, school children, and early shoppers. The peak travel and energy demand is reached at 9 A.M. The load rapidly diminishes as some of the trains return to the storage yards and the remainder carry fewer travelers. The minimum day load is reached during the noon hour and then rises continuously, increasing in rate of rise until the evening rush hour is in full swing with most of the workers homeward bound at 6 P.M. The load then falls rapidly until the rate of decrease is momentarily checked at about 10 P.M., when theatergoers, visitors, and the like, are homeward bound. After midnight the load again follows the cycle described.

Both the industrial and the traction loads are affected by the seasons in need for heating and lighting. Generally this will lift the entire curve for both these loads in the winter since the cold and darker days require the extra energy.

The load curves of long-distance electrified railroads are entirely different in shape from those for the urban roads. In fact some of the former have their peak loads in the early morning hours when freight-train movements reach their maximum as well as the overnight passenger-train movements. In general they do not tend to have the sharp double day peaks that the urban roads exhibit.

Load curves for dwellings (Fig. 30-2) are typical of a residential community rather than just one residence. During the early morning hours energy runs night lights, refrigerators, water heaters, oil burners, ventilating fans, and the like. In the winter the early risers require lights and, to prepare breakfast, toasters, percolators, waffle irons, etc. After the rising and breakfast hour the demand decreases somewhat, and equipment such as vacuum cleaners, radios, television sets, water heaters, and occasional lights makes a fairly constant demand for energy until about 4 P.M. Cooking appliances then cause a slight rise in demand.

After 4 P.M. the early sunset of winter brings the lights into action, and the total load rapidly approaches its peak about 5 P.M. during the month of December. The high level of demand persists until about 7 or 8 P.M., when the load drops fairly rapidly as the families retire or leave home to seek entertainment elsewhere, minimum loads being again reached at about 1 A.M. Loads of this class are generally less during the summertime principally because many families are away on vacation and cooking activity is at a minimum. The later sunsets cause a late hour for the peak demand, which is much reduced in comparison with the winter demand.

Street lighting is about the only form of load that does not exhibit peak demands in the ordinary sense. Normally all lights come on and are turned off almost simultaneously, the total load remaining constant during the hours of darkness. The turning on of lights and their turning off are usually synchronized, respectively, with the time of sunset and sunrise. Hence in the summertime the lights will be functioning for a much shorter time than during the winter. In some communities the lights may be turned off at some hour such as 11 P.M. or midnight and not turned on again until after the following sunset.

Typical metropolitan-area electric loads are also shown in Fig. 30-2 for a weekday, Saturday, and Sunday. These are typical for a large city requiring energy for industrial, traction, commercial, and residential needs from a common utility system. During the early morning hours the demand is at a minimum as the vast majority of the inhabitants sleep. The elements of this demand are the same as just discussed for each type of load curve. About 6 A.M. on the weekday, the load starts rising rapidly as the domestic and industrial loads increase as well as the traction load. By 10 A.M. the load reaches its maximum for the daylight hours. This is verified by the pronounced noon-hour dip. At 4 P.M. during the wintertime the load rapidly increases as the residential lighting load, the commercial load, and the traction loads increase on top of the basic industrial load. The total reaches a peak at about 5 P.M. and then rapidly decreases as the various components, excepting the residential load, decrease.

On Saturdays the noon-hour valley becomes an afternoon valley as some of the factories operate for only a half day so that the peak demand at about 6 P.M. is much less than the weekday peak.

Sundays are characterized by much lower demands in the absence of industrial demand and hence traction demand. During the daylight hours the load remains almost constant, rising only at the approach of darkness. The Sunday curve has a marked similarity to the residential load curve.

In the summertime the late sunsets occur after the industrial and traction loads are declining, with the result that the lighting demand is not coincident with these major components and the peak demand characteristic of wintertime does not appear.

Load curves reflect the activities of the population quite accurately in some instances. A late presidential broadcast or a major sporting event will increase the load above its usual value at that hour because more receivers and lights are in operation than normally. A cloudy or rainy day will be reflected in a higher than normal load during the day and evening compared with a day of clear weather. Thunderstorms during daylight hours invariably raise sharp peak demands of short duration caused by turning on lights.

The pronounced valley in the curves of the early morning hours is typical of all metropolitan-load curves in all seasons. To induce more use of generating equipment during these early hours, most utility systems have a preferential rate for users of energy who confine their major demands to this period and cease demand during the system peak-load hours. As will be evident after a study of incremental rates, any additional energy generated during these light-load periods can be provided at less than average cost up to a certain limit in load increase. Practical requirements of maintenance, boiler deslagging, and general cleaning of equipment can use the load decrease every 24 hr to permit shutting down some of the apparatus for these purposes.

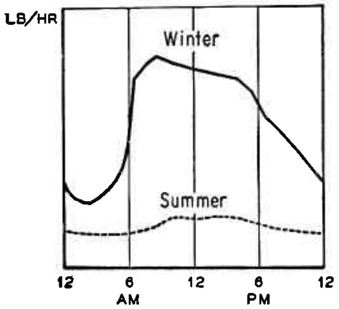

FIG. 30-3. Typical hourly variations in district steam-supply system.

30-4. Steam and Gas Loads. The weekday-load curves of a district steam plant supplying energy for space heating and process purposes are shown in Fig. 30-3. Shutting off heat supply entirely or holding a much lower temperature during the early morning hours causes a very deep valley in the load curve. However, about 5 A.M. heating systems come on in rapid succession to preheat buildings and apartments, with a peak demand about 8 to 9 A.M. The load then drops off as preheating ends and the outside temperatures warm up during the day. During the late afternoon the load drops rapidly as factories and offices close for the night. The summer curve lies far below the winter curve and represents mostly steam used for water heating and some process and power use.

A third type of common load curve in Fig. 30-4 is for gas supply in a metropolitan area. These curves are principally for domestic cooking, space heating, and commercial applications. In the winter curve three peaks are discernible, coiniciding with early morning space preheating and the preparation of breakfast, lunch, and dinner. The peak load for this system occurs on Thanksgiving Day, when the load peaks at the noon hour, coinciding with the traditional dinner hour of the feasting day.

30-5. Load-curve Analysis. The variation in demand throughout a period of time evidently may be considerable for a given plant, as demonstrated by the chronological load curves. It will be necessary for system planning and operating estimates to express the variation in, and the integration of, the total energy requirements for a period of time in some concise form; the load-duration curve does this.

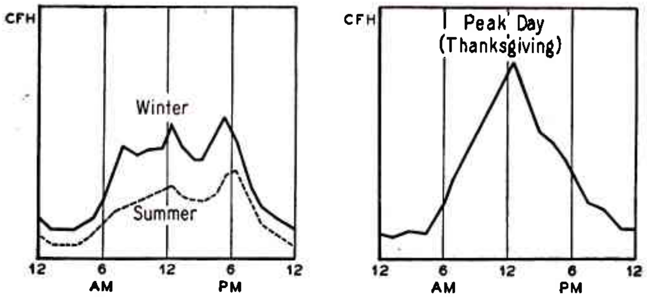

FIG. 30-4. Typical hourly variations in gas-supply system.

The area under the curve of a daily chronological load curve measures the total energy consumed by the load during the day. This energy is evaluated by $\int_0^{24} kw\, dt$, the unit being the kilowatthour. Fundamentally the load-duration curve is nothing more than a rearrangement of all the load elements of a chronological curve in the order of descending magnitude (Fig. 30-5). The areas under the load-duration and corresponding chronological curves are equal. Since it is impractical to determine the equation of either load curve, the area or energy is determined graphically. This can be done with sufficient accuracy in most cases by adding the readings off either curve at the mid-point for every half-hour period and dividing by 2.

Figure 30-5 shows the graphical method of constructing the load-duration curve from the chronological curve. The abscissa of the duration curve is laid off to equal the number of hours in the chronological curve, in this case 24 hr. The criterion of plotting the duration curve makes the abscissa at any load ordinate equal to the length of the abscissa intercepted by that load ordinate on the chronological curve. Thus, at the maximum demand or peak load, the intercept is one point which will be plotted at 0 hr. At load a the intercept is a_1 hr and is plotted at a_1 hr

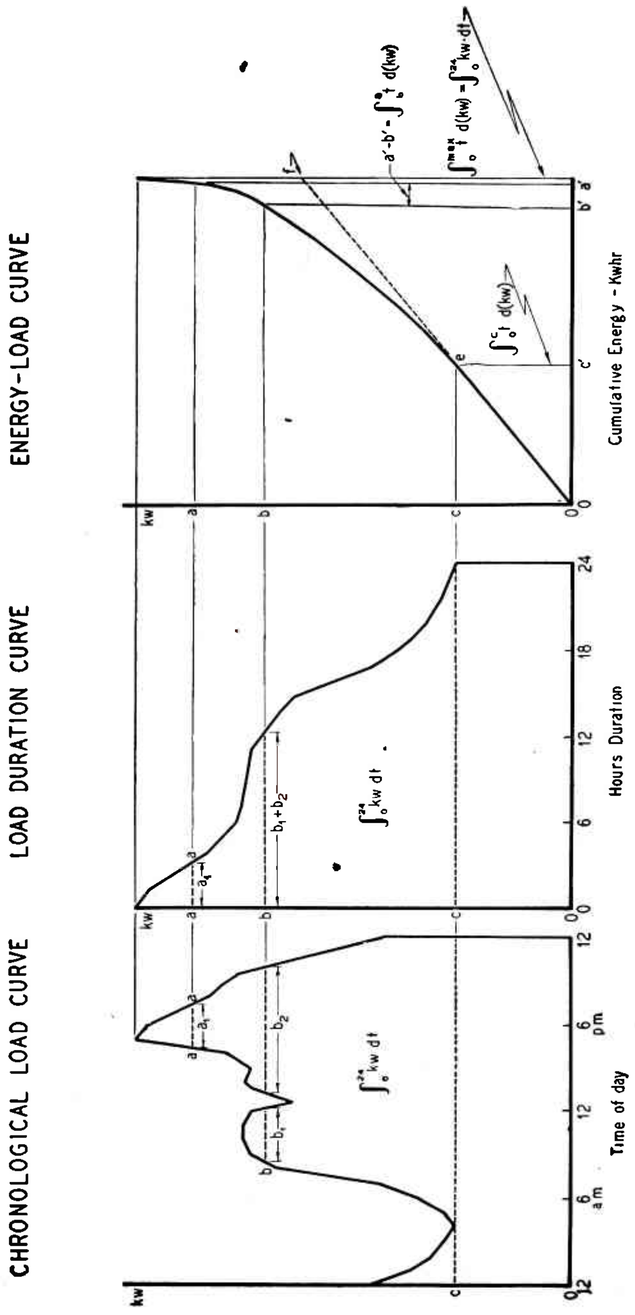

Fig. 30-5. Chronological load curve and its derived load-duration and energy-load curves.

on the duration curve. At load b the intercept is a total of $b_1 + b_2$ hr and is plotted accordingly. At minimum load c the intercept covers the entire period of 24 hr. Any point of the load-duration curve is a measure of the number of hours in a given period during which the given load and higher loads have prevailed. If the chronological curve indicated a constant demand during the entire day, it would be of rectangular shape and the load-duration curve would be an exact duplicate.

The duration curve can be extended to include any period of time. By laying out the abscissa to include 8,760 hr the variation and distribution in demand for an entire year can be summarized in one curve.

In power systems using hydraulic stations we must know the amount of energy between given demand levels on the load curve. This can be most easily found by plotting a load-energy curve which is derived from either the chronological or the duration curve. The load-energy curve (Fig. 30-5) plots the cumulative integration of the area under the load curve, starting at zero load vs. the particular load. Expressed mathematically, it is a plot of $\int_0^{kw_n} t\,d(kw)$ versus kw_n, where t is the hours of duration. Referring to the figure, the minimum load for the load curve is c kw. Between 0 and c kw, successive integrations are directly proportional to the load, and the load-energy curve plots as a straight line between these limits. For the next element of load above the minimum c kw, the elementary area is less than the elementary area at c kw. Hence the load-energy curve will deviate from a straight line and bend upward. Since the elementary areas subsequently decrease constantly as the kilowatt load increases, the curve has a continually increasing slope up to the peak load. Obviously the energy corresponding to the peak load will equal the total energy in the load curve. To find the energy included between the load values of a kw and b kw on the load curves, find the corresponding energies at a' and b' kwhr. The energy quantity will then be $a' - b'$ kwhr. Extending the straight-line portion to intersect the total kilowatthour abscissa determines the average load f. This intersection represents the load value that the chronological and duration curves would have if they were of rectangular shape and contained the same total energy.

In practice the true appearance of a load curve will not generally be so smooth as plotted in the illustrations. A pure lighting load will generally have the smoothest curve owing to the constant nature of the energy requirements of the individual lamps. At the other extreme the traction-load curve exhibits violent fluctuations in instantaneous values caused by the high and frequent starting demands of the trains, as well as the frequent stops requiring no energy. Some industrial loads also exhibit wide variations in instantaneous loads, especially where heavy stamping work or welding is performed. Central stations likewise experience momentary fluctuations in their total load, the extent of the variation naturally depending on the type of services supplied and also on the magnitude of the total load. In general, large systems have smaller fluctuations than smaller systems owing to the averaging effect of supplying a greater number of individual services.

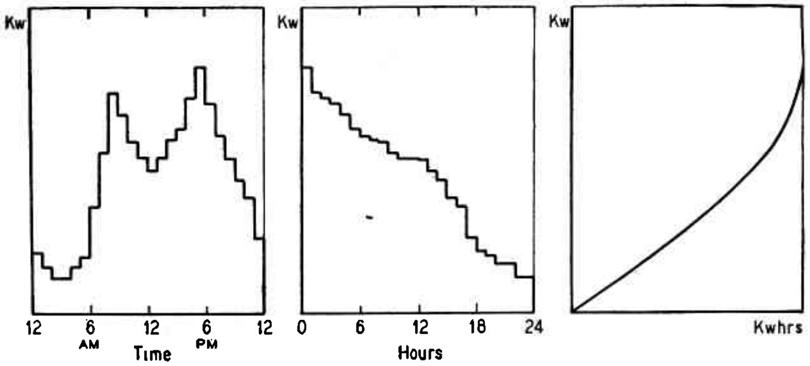

FIG. 30-6. Hourly-average-load curve and its derived load-duration and energy-load curves.

For loads exhibiting relatively minor variations in load the true load curve can be closely approximated by plotting the instantaneous readings taken every half or quarter hour and connecting the points with straight lines. It will usually be found that the area under such curves will generally be within 2 or 3 per cent of the total energy as measured by integrating meters. For loads of wide fluctuations the plotting of instantaneous readings is generally meaningless; the usual practice is to read the integrating meters every half or quarter hour and plot the difference in readings as an equivalent constant load for corresponding periods on the curve. This gives a step-shaped curve; at any instant this curve does not show the true load. It shows the trend of the load, and the area under the curve will exactly equal the total energy. Figure 30-6 illustrates the appearance of this type of chronological load curve with the corresponding load-duration curve and load-energy curve.

30-6. Load Factor. The degree of variation of the load over a period of time is measured by the load factor, which may be defined as

$$\text{Load factor} = \frac{L_{\text{avg}}}{L_{\max}} = \frac{E/h}{L_{\max}}$$

where L_{avg} = average load for period
$L_{\max}$ = peak load for period, either instantaneous or average, for periods of 1 hr or less
E = total energy in load curve for period
h = total number of hours in period

The load factor measures variation only and does not give any indication of the precise shape of the load-duration curve. Figure 30-7 shows the various shapes that load-duration curves may assume for a given maximum demand and load factor. As the load factor approaches zero, the duration curve will approach a narrow L shape, indicating a peak load of very short duration with very low or no load during the major portion of the time. As the load factor approaches unity, the duration curve will be somewhat rectangular in appearance, indicating high sustained loads.

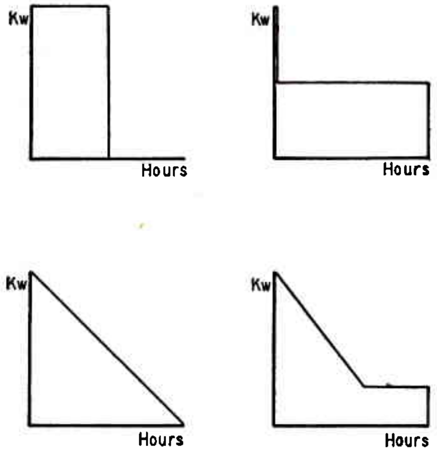

FIG. 30-7. Load-duration curves having common maximum demands and load factors.

In fact at unity the load curve can be only rectangular in shape, since maximum and average loads are equal.

30-7. Capacity Factor. The extent of use of the generating plant is measured by the capacity factor, frequently also termed *plant factor* or *use factor*. If during a given period a plant is kept fully loaded, it is evident that it is used to the maximum extent, or operated at 100 per cent capacity factor. If no energy was produced, the capacity factor would be 0 per cent. This factor is defined as

$$\text{Capacity factor} = \frac{L_{\text{avg}}}{Cap}$$

where *Cap* is the rated capacity of the plant.

Capacity factor and load factor become identical when the peak load is equal to the capacity of the plant. The relationship between the two factors is evidently

$$\text{Capacity factor} = \frac{L_{\max}}{Cap} \times \text{load factor}$$

30-8. Utilization Factor. The utilization factor measures the use made of the total installed capacity of the plant. The definition is

$$\text{Utilization factor} = \frac{L_{\max}}{Cap}$$

The factor for a plant depends upon the type of system of which it is a part. A low utilization factor may mean that the plant is used only for stand-by purposes on a system comprised of several stations or that capacity has been installed well in advance of need.

A high value of the factor, in the case of a plant in a large system, indicates that the plant is probably the most efficient in the system. In the case of isolated plants a high value means the likelihood of good design with some reserve-capacity allowance.

This factor may exceed unity, which indicates that loads have been carried in excess of the rated capacity of the equipment.

From the definition of the three factors in the foregoing discussion it will be evident that

$$\text{Capacity factor} = (\text{utilization factor}) \times (\text{load factor})$$

PROBLEMS

30-1. An electrical system experiences linear changes in load such that its daily load curve is described in Table P-1.

TABLE P-1

Time	Mw	Time	Mw
12 P.M.	20	12:30 P.M.	40
2 A.M.	10	1	50
6	10	5	50
8	50	6	70
12 M.	50	12	20

a. Plot the chronological and load-duration curve for the system.
b. Plot the load-energy curve for the system.
c. Find the load factor.
d. What is the utilization factor of the plant serving this load if its capacity is 100 mw?

30-2. An electrical railway system has such severely fluctuating loads that instantaneous wattmeter readings do not define its load curve adequately. The readings of Table P-2 have been taken from the station totalizing watthour meter at the times indicated. The station meter constant is 10,000 to convert to kilowatthours.

TABLE P-2

Time	Meter reading	Time	Meter reading
12 P.M.	5,595	2 P.M.	5,639
1 A.M.	5,597	3	5,643
6	5,602	4	5,648
7	5,605	5	5,654
8	5,611	6	5,661
9	5,618	7	5,667
10	5,624	8	5,672
11	5,629	9	5,676
12 M.	5,633	10	5,678
1 P.M.	5,636	12	5,682

a. Plot the average hourly chronological and load-duration curve.
b. Plot the load-energy curve for the average hourly loads.
c. Find the load factor based on the average hourly peak.
d. If the instantaneous peak is 85 mw, what is the load factor?
e. What is the utilization factor of the plant serving this load if its capacity is 100 mw?

30-3. An electrical system has linear variations in load so that the daily load curves can be described by the instantaneous readings of Table P-3. For a given week all the weekday load curves were identical.

TABLE P-3

Day or week	Time	Load, Mw	Time	Load, Mw
Weekday	12 P.M.	30	4 P.M.	60
	1 A.M.	20	5	70
	5	20	6	90
	6	40	8	80
	8	60	11	50
			12	30
Saturday	12 P.M.	30	12 M.	50
	1 A.M.	20	1 P.M.	40
	5	20	5	40
	6	40	6	60
	8	50	10	50
			12	30
Sunday	12 M.	30	5 P.M.	30
	1 A.M.	20	6	40
	8	20	10	40
	9	30	12	30

a. Plot the daily chronological curves on a common graph.
b. Plot the load-duration curve for the week.

30-4. The yearly duration curve of a certain plant can be considered as a straight line from 20,000 to 3,000 kw. To meet this load, three turbine-generator units, two rated at 10,000 kw each and one at 5,000 kw, are installed.

Determine:
a. Installed capacity
b. Plant factor
c. Maximum demand
d. Load factor
e. Utilization factor

30-5. Determine the maximum demand for the group of energy consumers shown in Table P-4.

TABLE P-4

Class of service	Total connected load, kw	Demand factor, per cent	Group diversity factor	Peak diversity factor
Public buildings	100	35	1.6	1.00
Apartments	1,000	55	4.0	1.20
Hospitals	200	45	1.5	1.05
Theaters	150	60	1.6	1.00
Laundries	50	70	1.8	1.05
Residences	3,000	40	4.5	1.20
Stores	500	65	1.6	1.05
Offices	100	70	1.8	1.05
Lighting (street)	600	100	1.0	1.00
Foundry	3,500	80	1.1	1.05
Boiler factory	4,000	90	1.1	1.05
Hotel	700	25	1.8	1.20
Motor factory	5,000	75	1.1	1.05

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CHAPTER 31

SELECTION OF PLANT

31-1. Plant Capacity. After defining the load expectations with a load-duration curve, the designer can determine the capacity of a plant. As a minimum requirement the plant capacity must be equal at least to the peak load, or maximum demand. Experience shows that no machine is so dependable that it may at all times be relied upon to be in operating condition when it is wanted, i.e., to be available for service 100 per cent of the time.

If an interruption of service can be tolerated at any time, then possibly the capacity equal to the peak load may be placed all in one unit of equipment, such as one diesel engine or one boiler feeding one turbine. However, if the load factor is comparatively low, it may be more economical to split the capacity among two or more units. Only a study of the particular circumstances can supply the correct solution.

In most cases an interruption of service cannot be tolerated. Perhaps the stoppage of some industrial process would entail costly spoilage; the cessation of some vital service, such as ventilation, might create a hazard to life. If so, provision must be made to maintain continuity of service within reasonable limits.

For a small load it may pay to install two units of equipment each being capable of supplying the maximum demand independently. In the event of failure of one unit the other will probably be available to carry the load. Though there is a possibility of both machines being unavailable simultaneously, it often will not justify the cost of installing a third reserve unit, since the probability of such an occurrence is rather remote. The remoteness of such a contingency is attested by the records of availability, on the average 90 per cent or better for equipment subjected to a regular inspection and overhaul schedule.

In some of the larger power-supply systems, made up of several generating units, design rules have been adopted to maintain a total installed capacity equal to the expected maximum demand plus the capacity of the two largest units. This is based on the expectation that at the time of the maximum demand one of the largest units or an equivalent amount of capacity may be out of service for overhauling.

Then, with all the other units actively operating, the peak load can still be carried if the other largest unit should fail owing to mechanical breakdown or operating error.

There is an inherent reserve in most electrical supply systems, in that the load can be decreased by dropping the voltage. A given percentage drop in voltage very nearly causes an equal percentage drop in load. In a large utility system a 5 per cent drop in load by this means usually equals a substantial-sized generating unit. By using this principle significant savings in fixed charges could be effected by resorting to a one-unit reserve plan in determining installed system capacity.

Most large units are conservatively rated. In the case of steam turbines the ultimate maximum capability is usually about 115 per cent of rated nameplate capacity. However, generators driven by these turbines may have a smaller ultimate capacity owing to internal temperatures at isolated points in the windings exceeding safe values. Nevertheless, during emergencies most individual units could produce more energy than their nameplate ratings indicate.

When an existing system prepares for an increase in maximum demand by installing an additional unit of generating equipment, the size of the unit will be based on three considerations: (1) the expected rate of increase of the maximum demand over a period of years, (2) the general design policy established by the system management, and (3) the room available for the additional unit. If the expected rate of rise is increasing and shows promise of continuing into the future, the capacity of the new unit may exceed the capacity of any existing unit with the expectancy of saving on total costs over a period of years. But if the rate of increase has been constant over several years with no expectation of change in the future, the new capacity very likely will equal that of the largest existing unit. The disadvantage of radical increases in unit capacities is the increase in the magnitude of reserve capacity required in following a given reserve policy.

31-2. Capacity Probability Analysis. With the steady growth of our larger utility systems with a multitude of units there is a need for a more sophisticated analysis of system capacity needs. In the past few years many utilities have used the probability theory in figuring their probable system outages to arrive at a suitable reserve capacity. Allowable space limits us here to just looking at the rudiments of the theory and its implications.

Let $Q_1, Q_2, Q_3, \ldots, Q_n$ be the forced outage rates of Unit No. 1, 2, 3, . . . , n, expressed as the ratio of number of days outage to the number of days in the year.

Similarly, let $P_1, P_2, P_3, \ldots, P_n$ be the operating rates of Unit No. 1, 2, 3, . . . , n, expressed as the ratio of number of days of running to the number of days in the year.

Then $P + Q = 1$ for any unit by definition.

From the probability theory the product P_1P_2 is the probability of Units No. 1 and 2 being in operation simultaneously over any given period; the product $P_1P_2P_3$ is the probability of Units No. 1, 2, and 3 being in operation simultaneously over any given period. The product $Q_1Q_2Q_3$ is the probability of Units No. 1, 2, and 3 being out of service simultaneously.

To find the probability of different combinations of units out of service and in service, we use

$$(P_1 + Q_1)(P_2 + Q_2)(P_3 + Q_3) \cdots (P_n + Q_n) = 1$$

If all the units have the same service and outage probabilities P and Q, this equation reduces to

$$(P + Q)^n = 1$$

Example 31-1. A generating system will have a total capacity of 300 mw; investigate the probable forced outages when the capacity is supplied in one unit; in two 150-mw units; in three 100-mw units; in four 75-mw units.

The operating probability of all units is $P = 0.98$, and the forced-outage probability rate is $Q = 0.02$.

For the single 300-mw unit the probability of having capacity *between* 0 and 300 mw out of service is 0.02.

For the two 150-mw units we must expand:

$$(P + Q)^2 = (0.98 + 0.02)^2 = 1$$
$$P^2 + 2PQ + Q^2 = 0.9604 + 0.0392 + 0.0004$$

The expression P^2 is the same as P_1P_2; the term $2PQ$ is the same as $P_1Q_2 + P_2Q_1$; and $Q^2 = Q_1Q_2$. The term P_1Q_2 is the probability of having Unit No. 1 in service and Unit No. 2 out of service simultaneously; the term P_2Q_1 is the converse.

Summarizing, we have the following tabulation:

Capacity out of service on forced outage, mw	*Probability*
0	0.9604
150	0.0392
300	0.0004
	1.0000

For the three 100-mw units we expand:

$$(P + Q)^3 = (0.98 + 0.02)^3 = 1$$

$$P^3 + 3P^2Q + 3PQ^2 + Q^3 = 0.941192 + 0.057624 + 0.001176 + 0.000008$$

Summarizing, we have the following tabulation:

Capacity out of service on forced outage, mw	*Probability*
0	0.941192
100	0.057624
200	0.001176
300	0.000008
	1.000000

For the four 75-mw units we expand:

$$(P+Q)^4 = (0.98+0.02)^4$$

$$P^4 + 4P^3Q + 6P^2Q^2 + 4PQ^3 + Q^4 = 0.92236816 + 0.07529536 + 0.00230496 + 0.00003136 + 0.00000016$$

Summarizing, we have the following tabulation:

Capacity out of service on forced outage, mw	*Probability*
0	0.92236816
75	0.07529536
150	0.00230496
225	0.00003136
300	0.00000016
	1.00000000

The probability of forced outages for the four schemes is plotted in Fig. 31-1. The plot shows two major principles that apply to units with equal probabilities: (1) the single unit has the best probability of maintaining maximum service in operation; (2) the four-unit system has the least probability of maximum outage.

For example, the probability of 0 mw loss (continuous operation) is 0.98 for the single 300-mw unit; this falls to 0.9224 for the four 75-mw units. This, however, is not the basic criterion for selecting a system for continuous operation. The probable outages are significant in this respect: The single unit is out of service 2 per cent of the time, but the four-unit system would have the same amount of capacity (300 mw) out of service only 0.000016 per cent of the time. Another way of stating this would be: All four units will be forced simultaneously out of service very seldom. The probability of forced outage decreases with an increasing number of units.

Figure 31-1 shows that the probability of capacity loss depends on the number of units and their capacity. Maximum outage of 1 day in 1 year would be 75 mw for the four-unit system, 100 mw for the three-unit system, and 150 mw for the two-unit system. The single-unit system cannot operate with this low an interruption rate. Maximum outage of 1 day in 16 years would be 150 mw for the four-unit system and 200 mw for the three-unit system.

In the various complete-analysis methods these outage probabilities are integrated with the system load curve to measure the total loss of kilowatthour generation. Lack of space prevents showing the technique.

Recent surveys have shown that the forced-outage rate for turbines using over 1,350-psig throttle steam averages at 0.89 per cent; their boilers have an average forced-outage rate of 0.90 per cent.

The service probability for each major plant element is:

Turbine:	1.0000 − 0.0089 = 0.9911
Boiler:	1.0000 − 0.0090 = 0.9910
Total unit:	0.9911 × 0.9910 = 0.98218

The corresponding forced-outage rate for the total unit is

$$1.0000 - 0.98218 = 0.01782$$

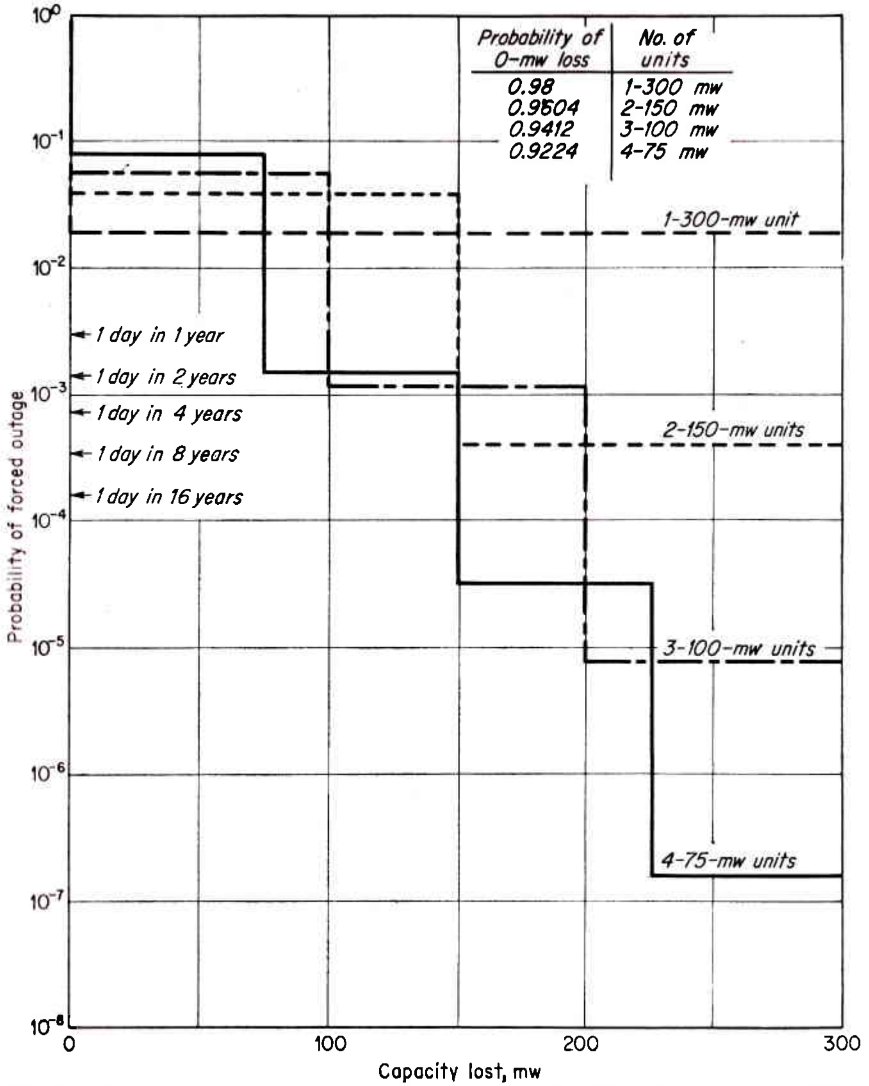

FIG. 31-1. Increasing capacity loss has a decreasing probability of occurrence. Using more than one unit decreases the probability of outages.

31-3. Plant Location. In providing energy supply exclusively to an industrial enterprise the choice of power-plant location is quite limited since almost invariably the plant must be located on or adjacent to the industrial property. If such a location fronts on a navigable river, lake, or sea, the site may be ideal for a condensing-steam plant as a source of energy supply. On the other hand, if water supply is limited and the cost of coal and oil is near a parity, the advantages of both a condensing-steam plant and an internal-combustion (i-c) plant using spray ponds or cooling towers should be investigated.

Should large amounts of steam be required in any of the manufacturing processes, large economies may be effected by bleeding a steam turbine as a source. The relatively high-temperature exhaust gases of a diesel engine or gas turbine are sometimes used to generate steam in a waste-heat boiler.

If the plant site is on a river with favorable topography and geological conditions and also adequate rate of water flow, a hydraulic installation may prove feasible. In any event all the possibilities should be surveyed and compared on a final total-annual-cost basis before an intelligent decision can be made as to the type of installation to be selected.

In contrast to the industrial installation the problem of plant location for a large regional utility system may be quite complex and may allow a wide range of choice in both location and type of installation. The type of plant at each possible location is dictated by the particular natural advantages of the site.

For thermal stations the plant location will be governed by joint consideration of: (1) transmission of energy, (2) fuel delivery, (3) water supply, and (4) realty value.

1. There will be considerable advantage in placing the source of energy supply as near the center of the load as possible. The saving in copper and transmission structures by reducing this distance will represent a saving in fixed costs within limits. The copper cross section directly depends upon the maximum current to be carried. For alternating currents this can be controlled by the choice of transmission voltage, since raising the voltage reduces the current and therefore the copper cross section for a given value of energy. However, increased voltage requires increased investment in insulation, which must be balanced against the decreasing costs of copper. The economic voltage usually increases with increase in transmission distance.

2. Fuel can be delivered in several different manners: by barge or ship if the plant is on a water front, by rail if along a railroad right of way, or by truck if neither of the other conveniences is available. The fewer times any commodity is handled in delivering to the ultimate consumer, the lower will be the cost of that commodity. It is important to choose a location that will bring this cost down to the economic minimum. The nearer the plant to the source of fuel, the less will be the cost of fuel to the user.

Fuel is often bought at the mine by the larger consumers, who then provide for its transportation to the point of consumption. The cost of transportation may finally represent as much as 80 per cent of the total cost of fuel to some consumers.

It is often proposed to locate principal power plants at the mouth of coal mines. Several plants are so located today. In many cases this is not practicable since the fixed cost of transmission facilities overbalances the saving in fuel transportation. The optimum plant location will be at some point where the sum of these two costs become a minimum, provided that other considerations are satisfied.

3. Availability of reasonably pure water supply as make-up is a requisite for a steam plant or, alternatively, a supply of water that is usable after suitable treatment. The latter is more costly, but most waters do require some kind of chemical treatment or distillation before introduction to the cycle.

An ample supply of cool water must be available for condenser cooling water in a steam plant. Since the cooling water increases in temperature as it absorbs energy from the condensing steam, it becomes important to limit this rise in temperature to the lowest practicable value. The condensing-steam temperature must be higher than the maximum cooling-water temperature to effect the necessary heat transfer. Thus, the larger the difference between initial cooling-water temperature and condensing-steam temperature, the lower the efficiency of the steam cycle and the lower the required mass of cooling water.

In many plants it is economical to limit the rise in cooling-water temperature to a small value to gain in cycle efficiency at the expense of increased cooling-water pumping requirements. The temperature rise usually falls between the limits of 10 and 20 F. Then, as there are usually between 950 and 1000 Btu in each pound of exhaust steam to be removed, the cooling-water requirements range from 45 to 100 lb per lb of steam condensed.

In the temperate latitudes, since the cooling water usually fluctuates with seasonal temperatures, a condensing-steam plant is more efficient in the winter than in the summer. To minimize this effect, the cooling-water pump speeds may be varied so that greater rates of water flow may be provided in the summertime.

For the large steam plant the large cooling-water requirement makes sites adjacent to large bodies of water preferable. But in arid areas many modern large plants use cooling towers. For the small plants that are distantly situated from rivers or lakes spray ponds or cooling towers may be employed. If feasible, wells may be driven to tap ground-water supply for cooling purposes. This usually suffices for jacket-cooling water in a diesel engine plant.

4. The cost of real estate upon which to construct a plant is relatively unpredictable. The price usually depends upon the probable competing uses for which the land may be used or wanted. Sometimes the mere hint that a large corporation is interested in a parcel of property may increase its price many times. The prices within city limits are high in comparison with a rural location. The fixed cost on the capital invested in real estate is not the only factor needing consideration. The taxes that will be levied at a location will be important in determining its desirability.

In studying alternate plant sites the final selection will be guided by the total annual costs that must be met for all the above items. In addition there may be other considerations that have to be taken into account, such as the expensive provisions for eliminating smoke and noise nuisance in a location near a residential district. These may be neglected in other situations.

The capacity of the plant will dictate the minimum size of plot and the quantity of fuel and water required, but otherwise it usually plays no part in deciding the final location of the plant. This decision is based almost wholly on the joint consideration and economic balance of energy transmission, fuel haulage, water supply, and realty cost.

31-4. Equipment Selection. With the capacity and location of the plant settled the designer must determine the specific energy equipment that will give him an economic plant. For a steam-turbine plant he will be confronted with the problem of making a selection of equipment with pressures ranging from 200 to 5,000 psi and temperatures from 500 to 1200 F. The equipment may be arranged in a simple Rankine cycle, a reheat cycle, or a regenerative cycle with any number of heaters that may be desired. Furthermore the station auxiliary equipment may be driven by steam turbines, engines, or electric motors. Finally there is a choice to be made between the products of different manufacturers.

The variation in theoretical efficiency of a cycle will be followed quite closely by the variation in actual thermal efficiency with given cycle conditions. The change in thermal efficiency for a simple Rankine cycle with a fixed ½-in.-Hg exhaust pressure with variation in throttle temperature and pressure is illustrated in Fig. 3-4.

The thermal advantage of increasing pressures and temperatures is obvious, but as the pressures increase, the thickness of metal and complication of construction required in boilers, prime movers, and piping increase, with a resulting increase in cost. As the temperature increases, the metallurgical specifications require rarer materials and the cost of the metal increases.

Aside from the vapor state at the throttle, increase in economy can be realized by installing heaters and auxiliary equipment to operate on the regenerative cycle. The increase in theoretical thermal efficiency with increase in the number of heaters is shown in Fig. 3-8. Each extra heater adds a smaller increment in efficiency than the preceding heater so that there is not much advantage in using a large number. The economic selection will be such that the net return on the heater investment will be as much as any alternate profitable investment that may be made with the funds expended on the heater equipment.

The i-c engine cycles become more efficient as the compression ratio rises. As the ratio increases, the maximum cylinder pressure increases and requires heavier cylinder walls, pistons, crankshafts, and general construction and, therefore, a more costly engine. A two-stroke-cycle engine is generally lighter in weight and less expensive than a four-stroke-cycle engine of the same capacity, but the latter will be more efficient thermally. Modern engines use superchargers to develop more power from a given cylinder at a higher thermal efficiency.

The thermodynamic cycle on which the engine operates determines the type of fuel that must be used, or more usually the kind of fuel available will determine the selection to be made. A plentiful and cheap supply of gasoline would act in favor of the Otto cycle; similar conditions for fuel oil would be in favor of a diesel engine. The latter condition is the case usually met in practice since fuel oil is generally cheaper than gasoline and both are usually available.

Diesel engines in practice have the advantage of operating at higher thermal efficiencies than comparable-sized steam plants, especially in the smaller capacity range. However, the maximum capacities of individual diesel engines are much smaller than for steam or hydraulic turbines. Maximum capacities of 30,000 kw in diesel engines have been installed in Europe, but 10,000 hp is the largest actually built in this country. The maximum capacity of steam turbines projected is 800,000 kw. Owing to this difference in capacities and aided by the fact that coal is usually cheaper than fuel oil in most localities, the diesel engine has not found much application in large regional power systems except as a temporary expedient. The i-c engine has found its place in small municipal energy systems and some commercial and industrial uses where steam is not required and where water is relatively scarce.

The selection of hydraulic-plant equipment depends principally upon the head of water available, in deciding among the three principal types: Pelton, Francis, and Kaplan. Obviously the hydraulic plant must be capable of meeting the maximum energy demand at the time of its occurrence. The rate of water flow in most practical situations seldom reaches a maximum at the time of the peak demand; i.e., there is a diversity between peak demand and peak rate of flow. As a result the usefulness of the plant site in an isolated system is limited in installed capacity to the probable water flow that will be available at the time of the peak demand, regardless of the fact that the flow may be many times the amount at some other time of the year. Most industrial hydraulic plants use only a fraction of the total flow in a river.

In a regional system containing several thermal plants the value of a hydraulic plant depends mostly upon the flow available at the time of the system's peak demand. The usual procedure measures the usefulness and economic possibilities of a proposed hydraulic station by comparing its total annual costs with those of an alternate steam plant having the same firm capacity. The firm capacity is the maximum generation that the hydraulic plant can produce at the time of the peak demand of the load curve of the system. The fuel cost in a large steam plant is often about one-half its total annual cost, the remaining costs being the fixed charges and the fairly constant cost components of operation. Since the operating cost of a hydraulic station is a small part of its total annual cost, it is economic to invest a greater amount initially, relative to a thermal plant, provided that the hydraulic plant can produce sufficient energy to serve the load at all the specific times when it is needed.

When the surrounding terrain permits, the amount of flow through a hydraulic plant can be increased above the natural flow rate by increasing the height of the dam and storing some of the river flow that would otherwise spill over it in off-peak hours. This stored water is termed "pondage." By drawing upon the pondage in addition to using the natural flow at time of the peak demand, the firm capacity of the plant can be increased.

Most hydraulic stations are supplemented by steam plants or diesel plants. The hydraulic plant often is only an incidental source of energy in a major power system. For this condition it often pays to install turbine capacity in excess of the capacity available in river flow at the time of the peak load. The installed turbine capacity may be equal to or less than the maximum river flow. Then, whenever water flow is of sufficient magnitude, the hydrogenerated energy will displace some generation at the thermal plants and save fuel.

Some natural sites for hydraulic plants are not developed owing to their distance from the nearest load center. The fixed charges on the investment necessary in transmission facilities added to the carrying costs of the plant, dam, riparian rights, and other items are enough to make thermal plants nearer the load center the better alternative scheme. There are other sites, ideal from a technical standpoint, that are impossible of development because the land that would be flooded is occupied by large towns or cities. The condemnation costs involved make the ventures economically prohibitive.

31-5. Equipment Cost. The cost of equipment varies with time, location, and capacity. Equipment prices are usually compared and quoted on the basis of price per unit of capacity, usually termed the "unit price." As the capacity of a given type of machine increases, the unit price decreases. Figure 31-2 shows the general trend and the trend of the major cost components in building a given type of machine. The material-cost curve decreases in slope as the capacity increases, starting from the origin. However, the engineering and labor costs, which include general overhead, increase only slightly with size or capacity. The labor and overhead expense to build a machine twice a given capacity is less than doubled in magnitude. The total-cost curve, which is the sum of the two foregoing components, has a positive intercept at zero capacity which represents the cost of just maintaining an organization of men and plant ready to produce. The shape of the total-cost curve is similar to the material-cost trend.

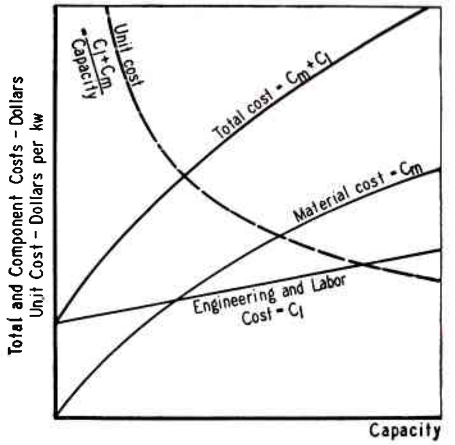

Fig. 31-2. Theoretical relations of component manufacturing costs, machine capacity, and cost per unit of capacity show lower unit cost for larger machines.

The reduction in unit price with increase in capacity is a major argument for installing large units of equipment. Evidently the cost of two machines equal in capacity to one large one will be more than the cost of the single large machine.

31-6. Plant Costs. Comparisons are made sometimes of different plants in respect to their relative installed unit costs. Direct numerical comparisons, however, are meaningless unless accompanied by a detailed analysis of each plant in respect to: the construction difficulties encountered during erection; special foundation needs; overtime work needed to meet operating deadline; delays due to storms or strikes; equipment-transportation costs; availability of materials and skilled labor; and other differences caused by location, political circumstances, and general financial conditions.

A comparison should also take into account the year during which a plant was built. Inflationary forces have steadily pushed prices upward, but skillful designing has counteracted this trend in large measure. Another factor lies in progressive manufacturing practices: after a new machine has been made available and has proved successful, its unit price tends to decrease. Development costs expended by the manufacturer are often recovered in the prices of the first few units built. After a number have been built, manufacturing economies are usually developed and are reflected in reduced prices for a better machine.

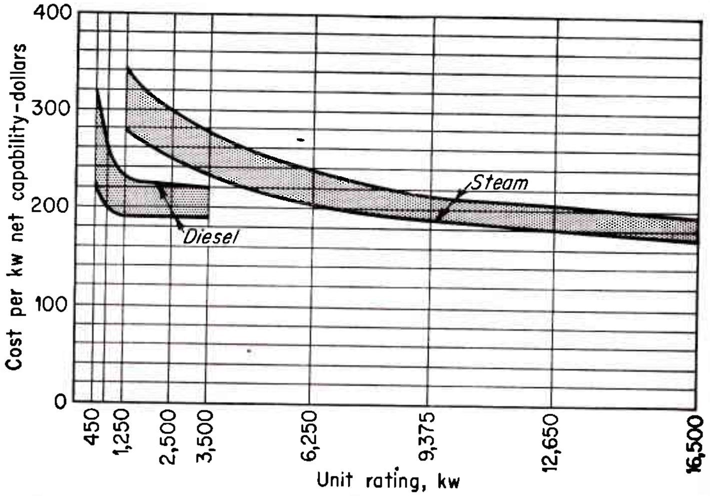

Fig. 31-3. Estimates of investments in small steam-electric and diesel-electric generating stations as of the mid-1950s. Lower limit indicates single-unit addition, while upper limit indicates unit in new plant.

Figure 31-3 shows one plant designer's estimate of the unit cost of small steam-electric and diesel-electric generating plants as of the mid-1950s. The upper curve applies to a new plant with one unit and the lower curve to the addition of a unit in an existing plant.

Figure 31-4 shows the unit costs for steam-electric central stations built during the period 1945 to 1957. The wide range in costs, even for equal capabilities, reflects the differing conditions under which they were built.

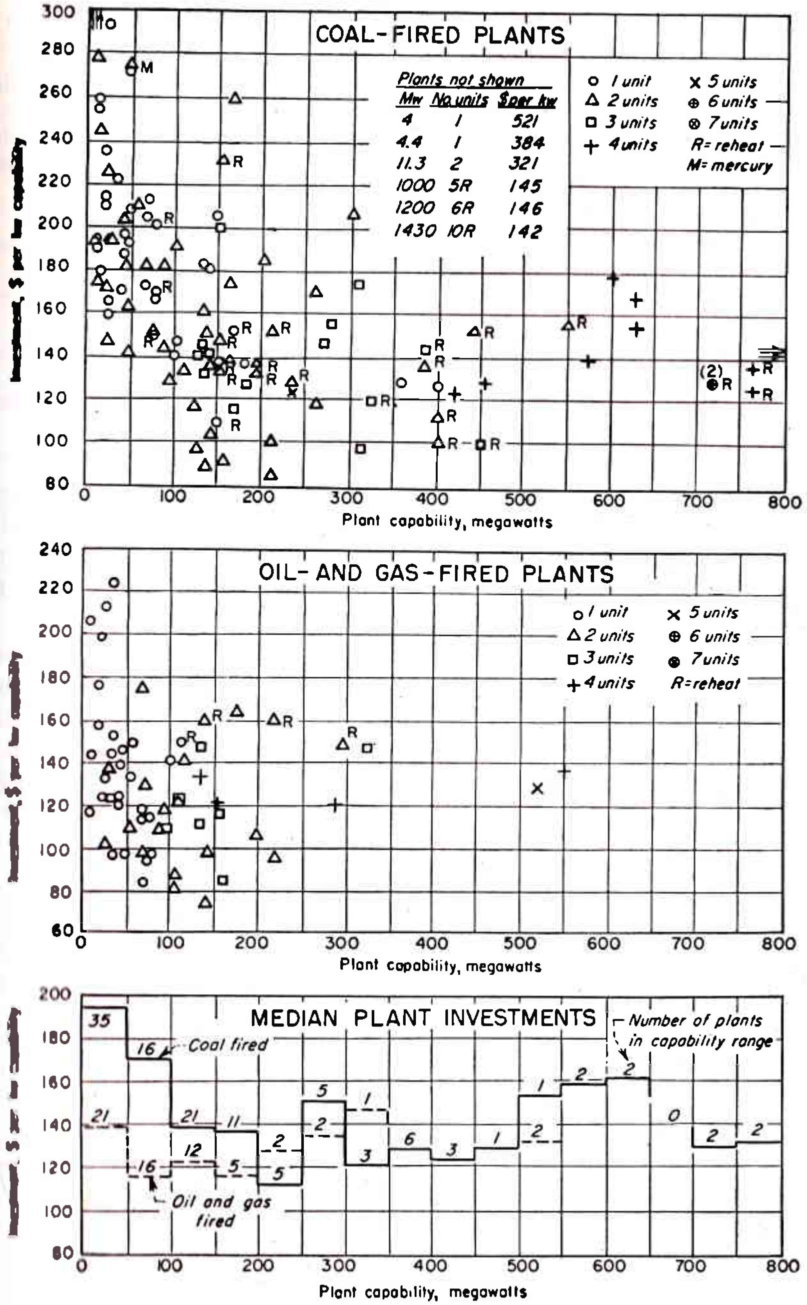

Fig. 31-4. Plot of unit investment costs for stations built between 1945 and 1957 shows wide range for all capabilities. In general oil- and gas-fired plants are lower than coal-fired plants, with some exceptions.

Samples of how low-cost and high-cost stations compare are given in the following tabulation:

Capability, mw	135	118	375
No. of units	1	1	2
Fuel fired	Coal	Gas	Coal, oil, gas
Investment, dollars per kw:			
Land	......	......	1.50
Structures	6.90	2.80	51.80
Boiler plant	52.20	35.00	79.10
Turbine generators	37.20	45.70	35.40
Electrical equipment	8.90	9.60	21.70
Miscellaneous equipment	0.80	0.90	1.70
Total	106.00	94.00	191.20

The unit cost of a plant cannot be taken as an unqualified measure of the excellence of design of a plant. The student should keep in mind that each plant is built to fit a unique set of conditions. Only a study of the economic situation surrounding the particular installation will show the quality of the engineering behind the construction of the plant.

Hydraulic-station costs are extremely variable in installed unit cost owing to the investment needed in dam and headworks, which may require a very small to a very large proportion of the total capital outlay, depending upon the natural features at the site. There is no point in quoting price ranges for given capacities because of the foregoing controlling conditions. Costs for most plants seem to range from $120 to $350 per kilowatt.

PROBLEMS

31-1. A central-station plant has the following turbogenerator units: two, 5,000 kw; four, 10,000 kw; and six, 25,000 kw.

a. If the installed reserve is required to be equal to the two largest units, what is the maximum protected peak this station may carry?

b. If the installed reserve is required to be equal to 10 per cent of the peak load, what is the maximum protected load this station may carry?

c. If this station is carrying a load of 156,300 kw at normal voltage and loses two of its largest units, what percentage drop in voltage is required to maintain operation and not increase the generation above the ratings of the remaining units?

31-2. The central station in Prob. 31-1 is now carrying a peak load of 150,000 kw. A new load is to be added to the lines and will add 30,000 kw to the peak demand. What is the minimum capacity that must be added to stay within an installed reserve requirement equal to the capacity of the two largest units? Can this capacity be added as one unit? Discuss.

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CHAPTER 32

STATION PERFORMANCE AND OPERATION CHARACTERISTICS

32-1. Performance Characteristics. Performance of generating plants is compared by their average thermal efficiencies over a period of time. Average thermal efficiency is the ratio of useful energy output during the period to the total energy input for the period. This measure of performance varies with uncontrollable conditions such as cooling-water temperature, shape of load-duration curve, total output, and quality of fuel; hence it is not a satisfactory standard of comparison unless all plant performances are corrected to the same controlling conditions. Such corrections are difficult to establish and questionable as to their accuracy.

Plant performance is most precisely described by the input-output curve derived from tests of the individual equipment. Figure 32-1*a* shows the general trend of such a curve, which follows the form defined by

$$y = a + bx + cx^2 + dx^3 + \cdots + nx^r$$

The time consumed in finding the equation of an input-output curve usually cannot be justified, and so the curve is always expressed graphically. In this text, however, since certain relationships between the input-output curve and its derived curves can be more clearly demonstrated mathematically and because of the amount of space required by curves to be read to four significant figures, the characteristics will be given mathematically for examples and problem work.

At zero load L the positive intercept for I measures the amount of energy required to keep the apparatus functioning. This energy dissipates as frictional and heat losses. Any additional input over the no-load input produces a certain output, the magnitude depending upon the machine. All the additional input does not appear as output, owing to partial dissipation as losses. From the basic input-output curve the more familiar efficiency curve and heat-rate curve may be derived directly, and also the incremental-rate curve. The use of the latter will be demonstrated later. An example of these companion curves is given in Fig. 32-1*b* and *c*.

For the steam turbine, steam generator, steam station as a whole, or diesel engine and station the input I is expressed in millions of Btu per hour and the output or load L in kilowatts. In a hydraulic plant the input would be in terms of cubic feet per second of water flow and the load in kilowatts.

The efficiency curve is derived by taking at each load the corresponding input; then

$$e = \frac{3{,}413\,L}{I} \times 100 \qquad \text{per cent}$$

is plotted against the value of L. Instead of expressing the efficiency

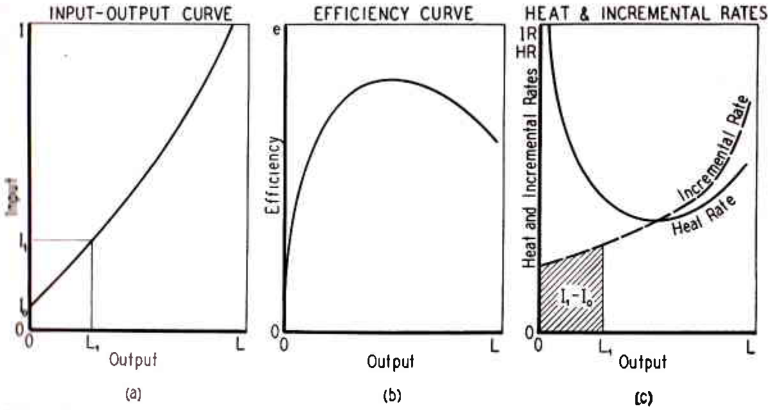


FIG. 32-1. Input-output curve and corresponding efficiency, heat-rate, and incremental-rate curves.

as a percentage it could be given as a ratio, the unit then being the decimal part of a unit of output per unit of input.

The heat-rate curve is derived by taking at each load the corresponding input; then

$$HR = \frac{I}{L} \qquad \text{Btu per kwhr}$$

is plotted against the corresponding value of L. The heat rate is a direct function of the reciprocal of the efficiency since

$$HR = \frac{I}{L} = \frac{1}{L/I} = \frac{1}{e/(3{,}413 \times 100)} = \frac{341{,}300}{e}$$

The heat-rate curve can be expressed also mathematically. Thus, if the input-output curve is defined by

$$I = a + bL + cL^2 + dL^3$$

then

$$HR = \frac{I}{L} = \frac{a}{L} + b + cL + dL^2$$

The incremental-rate curve is derived from the input-output curve by finding at any load the additional (or differential) input required for a given additional (or differential) load; the incremental rate defined as

$$IR = \frac{dI}{dL} \qquad \text{Btu per kwhr}$$

is plotted vs. the load L for the full range of the input-output curve. Mathematically the incremental rate is the slope of the input-output curve at the given load. Physically the incremental rate expresses the amount of additional energy required to produce an added unit of output at any given load.

In Fig. 32-1 areas under the input-output curve have no significance; on the other hand, from the definition of an incremental rate, the areas under the incremental-rate curve measure the additional input required to increase output from L_1 to L_2; since

$$dI = IR\,dL$$

then

$$I_2 - I_1 = \int_{L_1}^{L_2} IR\,dL$$

In the figure the area under the incremental-rate curve from $L = 0$ to L_1 is equal to the difference $I_1 - I_0$ on the input-output curve.

If the input-output curve is defined by

$$I = a + bL + cL^2 + dL^3$$

then the corresponding incremental-rate curve is defined by

$$IR = \frac{dI}{dL} = b + 2cL + 3\,dL^2$$

Example 32-1. A 20-mw station has an input-output curve defined by

$$I = 30 + 0.5L + 0.65L^2 + 0.01L^3$$

where I is in millions of Btu per hour and L is in megawatts.

Find the increase in input required to increase the station output from 7 to 9 mw by means of the input-output curve and also by the incremental-rate curve.

Solution. From the input-output curve

At $L = 7$:

$$I = 30 + (0.5 \times 7) + (0.65 \times 49) + (0.01 \times 343) = 68.78 \times 10^6 \text{ Btu per hr}$$

At $L = 9$:

$$I = 30 + (0.5 \times 9) + (0.65 \times 81) + (0.01 \times 729) = 94.44 \times 10^6 \text{ Btu per hr}$$

Increment input for 2-mw increased output = (94.44 − 68.78)10^6
= 25.66 × 10^6 Btu per hr

From the incremental-rate curve

$$IR = \frac{dI}{dL} = 0.5 + 1.3L + 0.03L^2$$

Assuming that from $L = 7$ to $L = 9$ the incremental-rate curve may be considered to be a straight line without serious error, then the average height of area under the curve between these two loads would be the incremental rate at $L = 8$ mw, or

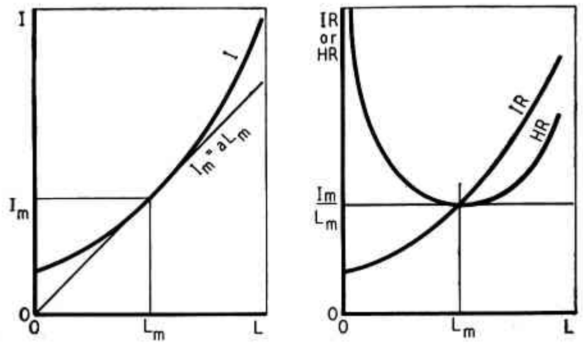
$$IR = 0.5 + (1.3 \times 8) + (0.03 \times 64) = 12.82 \times 10^6 \text{ Btu per mwhr}$$

Then the increment input for 2 mw increased output is

$$2 \times 12.82 \times 10^6 = 25.64 \times 10^6 \text{ Btu per hr}$$

(cf. 25.66 × 10^6 Btu per hr).

This good agreement between the two answers follows from the validity of the assumption made regarding the close approximation to a straight line of the incremental-rate curve for the relatively small increase in load. For a large increment of load this assumption would not hold because of the marked curvature of the incremental-rate curve; for such a condition to approximate closely the true increment input, the total area would have to be arbitrarily divided into smaller areas and the height of each of these determined. The application of this method of determining increment inputs will be demonstrated in a later chapter. The reader is urged to plot the curves of this problem to afford a more complete appreciation of the discussion.



(a) (b)

FIG. 32-2. Curves demonstrating equality of heat and incremental rates at minimum heat rate for given input-output curve.

Note in Fig. 32-1c that the incremental-rate curve crosses the heat-rate curve at the lowest value of heat rate when the two curves are plotted on common coordinates. The reason for this property becomes evident upon the following analysis:

Figure 32-2a represents a continuous input-output curve on which there will be one point (I_m, L_m) where the tangent to the curve will pass through the origin. The equation of this tangent is

$$I_m = aL_m$$

where a is the slope of the tangent. But at L_m the heat rate is

$$HR_m = \frac{I_m}{L_m} = a$$

Now, if the tangent were considered an input-output curve, the corresponding heat-rate curve and incremental-rate curve would be a horizontal line with a value of I_m/L_m, as shown in Fig. 32-2*b*. Since the actual input-output curve lies above this tangent at *all* other loads than L_m, it follows that the heat-rate curve must be higher than I_m/L_m at *all* other loads. Then the heat rate is a minimum when $HR_m = a$.

The slope of the tangent to the actual input-output curve at L_m, being the slope of the curve at L_m, therefore equals the incremental rate at this load, or

$$a = IR_m = \frac{dI_m}{dL_m} \qquad \text{at } L_m$$

But
$$a = HR_m \qquad \text{at } L_m$$

Then
$$IR_m = HR_m$$

or the heat rate of a continuous input-output curve is at a minimum when it equals the incremental rate. Since the incremental-rate curve has a continuous increasing characteristic, the two curves must cross at L_m when $HR_m = IR_m$. A consideration of Fig. 32-2*b* shows that, for any load between zero and L_m,

$$IR < HR$$

or
$$\frac{dI}{dL} < \frac{I}{L}$$

Then within this load range for any increase in load from L to $L + dL$ the heat rate decreases since

$$\frac{I}{L} > \frac{I + dI}{L + dL}$$

Conversely, for loads in excess of L_m,

$$IR > HR$$

or
$$\frac{dI}{dL} > \frac{I}{L}$$

and it follows that for an increase in load from L to $L + dL$ the heat rate increases since

$$\frac{I}{L} < \frac{I + dI}{L + dL}$$

In other words, starting from zero load, the relatively low rate of additional input per unit of additional output helps to decrease the heat rate as the load increases until the heat rate equals the incremental rate. After the minimum heat rate is reached, the increment rate, which is continuously increasing, causes a subsequent continuous rise in heat rate.

Equality of heat rate and incremental rate of an input-output curve at minimum heat rate is demonstrated very briefly by the calculus as follows:

At minimum heat rate the slope of the heat-rate curve is zero,

$$\frac{d(HR)}{dL} = \frac{d(I/L)}{dL} = 0$$

or
$$\frac{L\,dI - I\,dL}{L^2} = 0$$

Then
$$L\,dI = I\,dL$$

and
$$\frac{dI}{dL} = \frac{I}{L}$$

or
$$IR = HR \qquad \text{when } HR \text{ is a minimum}$$

32-2. Origin of Performance Data. When new equipment for installation is being considered, the manufacturer will supply the necessary performance data. Such data are derived from design calculations and numerous experimental and shop tests that were run by the manufacturer on similar equipment. For turbines manufacturers often quote expected performance from one-quarter to full load and guarantee a performance 1 per cent poorer than expected. For steam generators it is customary to guarantee the performance only at full load. Though performance at other loads may be quoted, it is not guaranteed.

Input-output and associated characteristics of existing equipment are found by tests run in the plant. Considerable care and preparation are required to ensure results of a reasonable order of accuracy. The method of testing power plant equipment is codified in minute detail in the power test codes sponsored by the ASME. At best, tests of the highest order of precision are only approximations because measurements of physical quantities are in reality approximations. The accuracy of tests or measurements depends upon the following factors: (1) the precision of instruments used for measurements, (2) proper application of instruments to ensure correct indications, (3) exactness of reading of instrument indications by observers, (4) degree to which a sample of a batch truly represents the entire batch, and (5) the method of conducting and computing the test and test results.

The precision of tests of turbines is usually of the order of ±2 per cent; for stoker-fired steam generators it is about ±5 per cent; for steam generators fired by gas, oil, and pulverized coal it is about ±4 per cent.

Turbines are tested by making runs for at least 1 hr at each maximum valve opening and at loads halfway between the maximum valve openings.

Steam flow is condensed and weighed; generator outputs are read by indicating wattmeters every minute; and temperatures, pressures, vacuums, and barometers are read every 5 min. All instruments are calibrated against suitable standards before and after the test, and corrections determined from these calibrations are applied to the observed data before making final calculations. It is customary to correct the calculated test performance to standard conditions of pressure and temperature at the throttle and standard pressure at the exhaust. If the turbine drives an a-c generator, corrections also must be made to a standard power factor. Where it is impossible to condense the steam, the test code permits use of standard orifice plates to measure the fluid flow, the manometers being read every minute during the test run. If the fluctuations are very severe, the test run may be increased to 2 hr or more to reduce the influence of the errors of manometer reading.

Duration of steam-generator test runs depends upon the uncertainty in fuel measurements and water measurements; for instance, in a stoker-fired steam generator there is always a certain amount of incompletely burned fuel on the stoker, and it is extremely difficult to make this the same at the beginning and end of a test run. Similarly the water level in the boiler drum may be different at the beginning and end of a test run even though the gauge may read the same. By making a test run long enough the effect of these unknown quantities can be reduced to a suitable degree. Usually low-load test runs are made for 48-hr and full-load runs for about 24 hr. Pulverized-coal-fired and oil- and gas-fired boilers are generally run 8 to 12 hr on each run. Feedwater to the boiler is weighed, as well as the fuel; samples of fuel are taken from each weighed batch for calorific and ultimate analyses; temperatures, pressures, and drafts are observed every 15 or 30 min; and gas analyses are made at various passes in the steam generator every 15 or 30 min.

Two or more test runs are usually made at each of four or more rates of steam generation. It will be evident that determination of the boiler input-output curve involves much more work and time than the comparable turbine characteristic. However, it requires probably more skill to operate a steam generator at optimum efficiency at all times than a steam turbine or steam engine.

32-3. Average, or Integrated, Heat Rate. With a load-duration curve and input-output curve given we can predict the average heat rate of the operation of a unit of equipment or a plant as a whole for the given period, since

$$\text{Average } HR = \frac{\text{total input during period}}{\text{total output during period}}$$

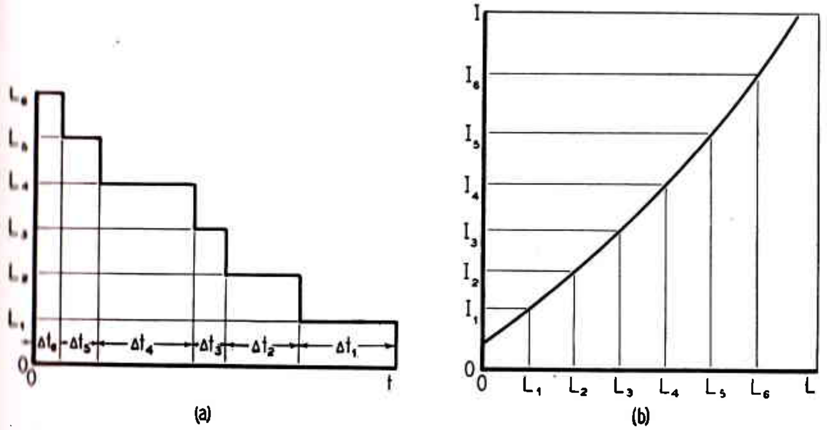
The average heat rate (see Fig. 32-3) for a plant with the input-output characteristic shown operating on the given load-duration curve would be

$$\text{Average } HR = \frac{I_1\,\Delta t_1 + I_2\,\Delta t_2 + \cdots + I_6\,\Delta t_6}{L_1\,\Delta t_1 + L_2\,\Delta t_2 + \cdots + L_6\,\Delta t_6}$$

As an alternate evaluation, divide both numerator and denominator by the total time t; then

$$\text{Average } HR = \frac{\Sigma(I\,\Delta t)/t}{\Sigma(L\,\Delta t)/t} = \frac{I'}{L'} = \frac{\text{average input rate}}{\text{average load}}$$

where $t = \Sigma(\Delta t)$ is the total hours in the period.



(a) (b)

Fig. 32-3 Application of load-duration curve to input-output curve in determining average heat rate.

The load-duration curve, as shown by the solid line in Fig. 32-4, is a typical shape for an actual plant. This shape proves troublesome in computing the average heat rate. It may be simplified to an equivalent stepped curve by dividing the total time into a convenient number of intervals. The average load for each interval is found, usually by inspection, by keeping the area under the average load curve (horizontal dotted line) equal to the area under the actual load curve (solid line). Careful investigation shows that this modification introduces an entirely negligible error and is completely justified by the saving in calculating time.

Despite the fact that the input-output curve is a plot of *instantaneous* input rate vs. the *instantaneous* load, the same numerical coordinates can be used to plot the average input rates vs. the average loads. For the general shape of load-duration curve, the average load vs. the average input will plot at some point above the input-output curve. That all possible average performance points will lie within a certain area above the input-output curve is the purpose of the following demonstration and proof:

In Fig. 32-5*a* and *b* are shown an input-output curve and its corresponding heat-rate curve. If the station should operate at a constant load L_a in Fig. 32-5*c* for a total considered period, the average input rate and average load will be identical, respectively, with the instantaneous input rate and load. Then, for a series of constant-load load-duration curves as in Fig. 32-5*c*, the average input-output curve **1-*a*-*b*-2** as in Fig. 32-5*e* would be a reproduction of the instantaneous curve **1-*a*-*b*-2** in Fig. 32-5*a*. The same holds for the instantaneous and average heat-rate curves as in Fig. 32-5*b* and *f*.

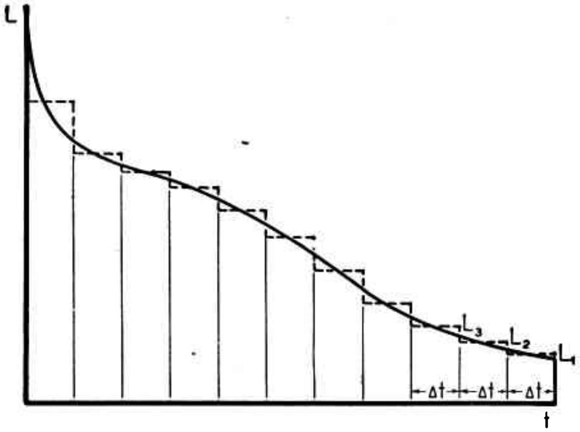


Fig. 32-4. Method of simplifying a load-duration curve.

Suppose that the station operates at only two different loads during an entire period such as at L_a and L_b in Fig. 32-5*d*. Then, as the number of hours of operation at each load is varied, the load-duration-curve shapes would vary as indicated by the dotted lines in the figure. The locus of the average heat rates in operating at the two load levels exclusively can be demonstrated as follows:

Let

I' = average input rate for total period of t hr
L' = average load for total period of t hr
I_a = instantaneous input rate for instantaneous output of L_a
I_b = instantaneous input rate for instantaneous output of L_b
t_a = hours of operation at load L_a
t_b = hours of operation at load L_b

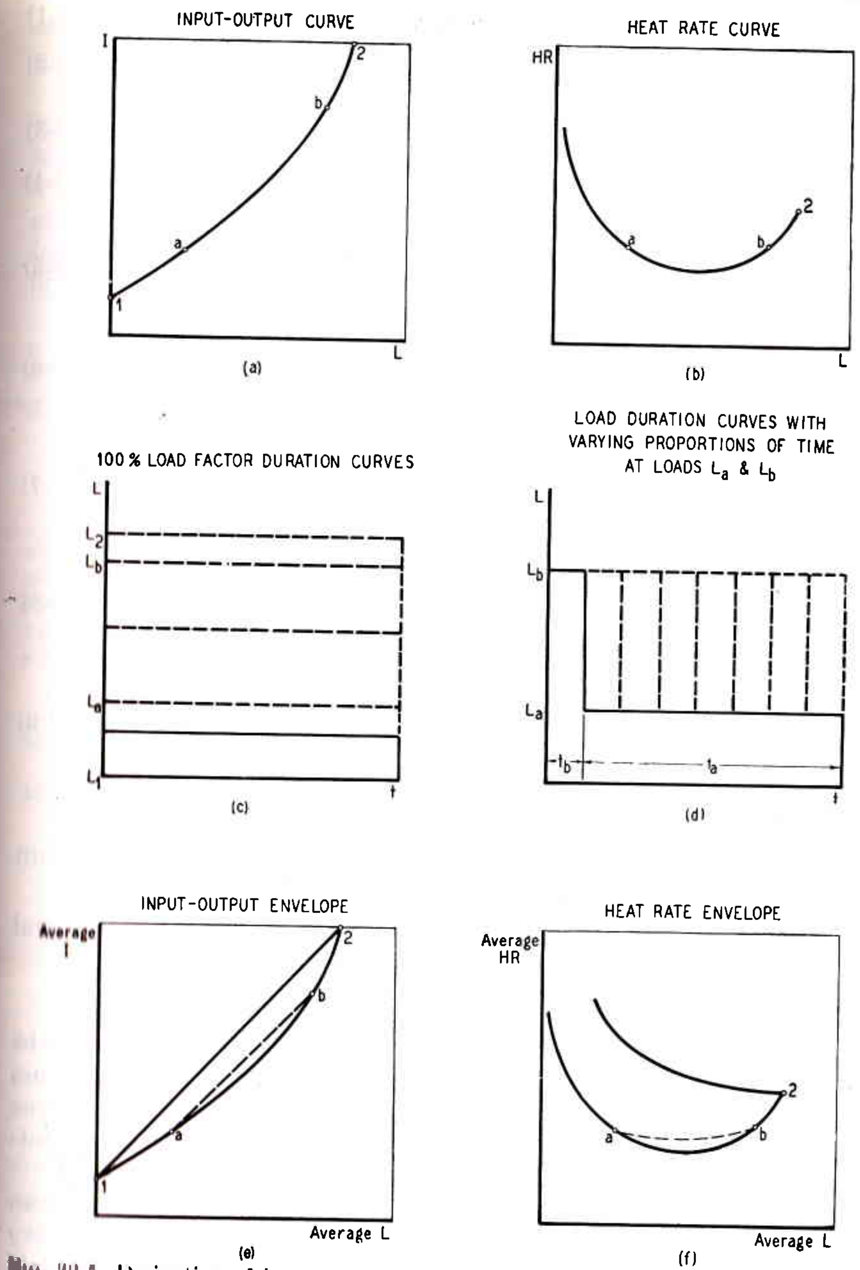


Fig. 32-5. Derivation of input-output and heat-rate envelopes of average values.

Then

$$t = t_a + t_b \tag{32-1}$$

$$I' = \frac{I_a t_a + I_b t_b}{t} \tag{32-2}$$

$$L' = \frac{L_b t_b + L_a t_a}{t} \tag{32-3}$$

From (32-1)

$$t_a = t - t_b \tag{32-4}$$

Substituting in (32-2),

$$I' = \frac{I_a(t - t_b) + I_b t_b}{t} \tag{32-5}$$

and solving,

$$t = \frac{t_b(I_b - I_a)}{I' - I_a} \tag{32-6}$$

Substituting (32-4) in (32-3),

$$L' = \frac{L_a(t - t_b) + L_b t_b}{t} \tag{32-7}$$

and solving,

$$t = \frac{t_b(L_b - L_a)}{L' - L_a} \tag{32-8}$$

Equating (32-6) and (32-8),

$$\frac{t_b(I_b - I_a)}{I' - I_a} = \frac{t_b(L_b - L_a)}{L' - L_a} \tag{32-9}$$

Clearing and transposing,

$$I' = \left[I_a - \frac{L_a(I_b - I_a)}{L_b - L_a}\right] + \left[\frac{I_b - I_a}{L_b - L_a}\right] L' \tag{32-10}$$

All the factors in the bracketed terms are constants. Then the general form of the equation is

$$y = a + bx$$

where I' and L' correspond to the variables. Then the locus of the average input-output points for varying hours of operation at two values of loading, keeping the total period hours constant, is a straight line connecting the two points (corresponding to the load values) on the input-output characteristic such as the dotted line *a-b* in Fig. 32-5*e*.

By inspection, it will be seen that a straight line connecting the two terminal points of the curve 1-2 will be above any line connecting *any* two points on the curve. Furthermore, no point defined by I' and L' can be below the curve 1-*a*-*b*-2. Then the area 1-*a*-*b*-2-1 constitutes an *envelope* that encompasses all possible values of average input and average output intersections or plots. The corresponding heat-rate envelope is shown in Fig. 32-5*f*. The left end of the latter envelope is at zero load and infinite heat rate.

Thus, depending upon the degree of curvature of the basic *I-L* curve, a power station may have a wide or narrow range of possible average heat rates. At a given average load it will, therefore, be possible to have a considerable variation in average heat rate, the magnitude being governed by the shape of the load-duration curve. The limiting case for zero envelope area occurs when the input-output curve is a straight line.

The envelope area also diminishes as the limits of load between which a load curve lies is narrowed, such as between loads L_a and L_b in Fig. 32-5*e* and *f*. The upper limit of average input rate is defined by the dotted curves drawn between *a* and *b*.

Example 32-2. The input-output curve of a 20-mw generating station is defined by

$$I = 30 + 0.5L + 0.65L^2$$

where I is in millions of Btu per hour and L is in megawatts. Find the average heat rate of this station for a day when it was operating at a load of 20 mw for 12 hr and was kept hot at zero load for the remaining 12 hr. Compare this average heat rate with the heat rate that would obtain if the same energy were produced for the day at a constant 24-hr load, i.e., at 100 per cent load factor.

Solution

At $L = 0$:

$$I = 30 + 0.5 \times 0 + 0.65 \times 0 = 30 \times 10^6 \text{ Btu per hr}$$

At $L = 20$:

$$I = 30 + 0.5 \times 20 + 0.65 \times 400 = 300 \times 10^6 \text{ Btu per hr}$$

$$\Sigma L = (0 \times 12) + (20 \times 12) = 240 \text{ mwhr}$$

$$\Sigma I = (30 \times 12) + (300 \times 12) = 3{,}960 \times 10^6 \text{ Btu}$$

$$\text{Average daily } HR = \frac{3{,}960 \times 10^6}{240 \times 10^3} = 16{,}500 \text{ Btu per kwhr}$$

$$\text{Average } L = {}^{240}\!/_{24} = 10 \text{ mw}$$

$$HR = \frac{30}{L} + 0.5 + 0.65L$$

At $L = 10$:

$$HR = 3.0 + 0.5 + 6.5 = 10 \times 10^6 \text{ Btu per mwhr, or } 10{,}000 \text{ Btu per kwhr}$$

Thus, if it were possible to redistribute the generation of the total daily output to a constant rate for the 24 hr, a saving of

$$16{,}500 - 10{,}000 = 6{,}500$$

Btu per kwhr could be achieved.

The input-output curve and its companion curves can just as readily be plotted with the capacity factor as abscissa instead of the kilowatt load. By plotting in this fashion it can be more readily demonstrated that a narrower envelope defined by the load factor will encompass possible performance points.

As the first step of this analysis note that the range of values that the load factor can assume is definitely limited by the value of the capacity factor. It has been shown in Chap. 30 that

$$LF = CF \times \frac{Cap}{L_{\max}}$$

where LF = load factor
CF = capacity factor
Cap = capacity, kw
$L_{\max}$ = peak load, kw

Then for the normal case, since capacity is a constant and the peak load cannot exceed the capacity, the load factor for a given value of capacity factor can assume any value between unity as an upper limit and the magnitude of the capacity factor as a lower limit (Fig. 32-6).

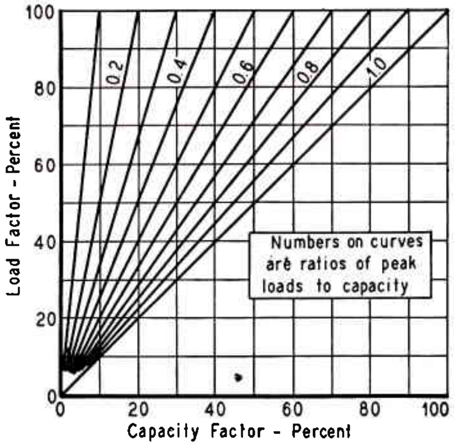


FIG. 32-6. Correlation of load and capacity factors.

In the input-output curve in Fig. 32-7 the abscissa is scaled in terms of capacity factor. Assuming that the equipment represented by this curve operates between loads equivalent to 10 and 100 per cent capacity factor, the over-all envelope can be immediately drawn by connecting these two extreme values by a straight line. Now at any given capacity factor the upper limit of average input will be on the straight line which will be at the lowest possible value of load factor (= capacity factor). It has been shown in a previous demonstration that the lower limit of average input rate will fall on the input-output curve, for which condition the load factor is unity, or 100 per cent.

At any selected intermediate value of load factor for a given magnitude of capacity factor, the upper limit of the average input will be found by determining the value of the corresponding maximum load as a percentage or ratio of the capacity, i.e.,

$$\frac{L_{\max}}{Cap} = \frac{CF}{LF}$$

This maximum load is located on the input-output curve by regarding it as an equivalent capacity factor. For example, referring to the figure, if at 60 per cent capacity factor the upper limit of average input is wanted for a load factor of 75 per cent, then

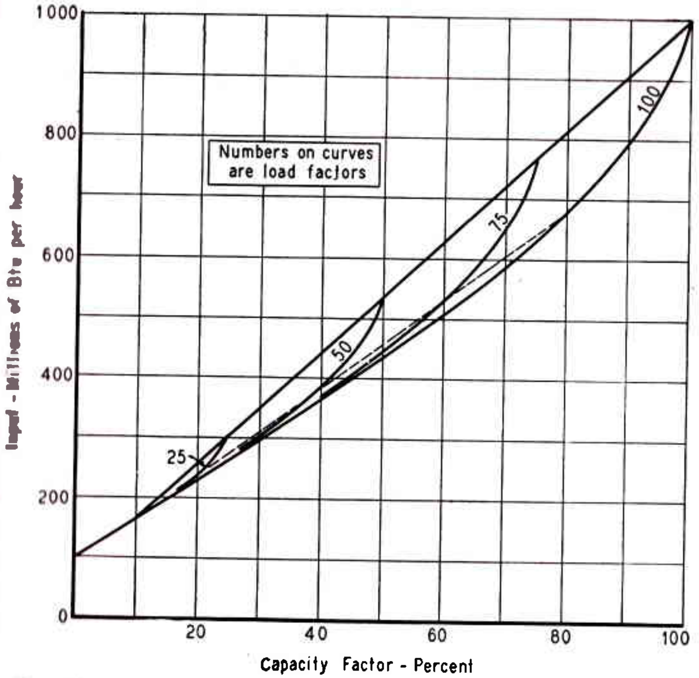


FIG. 32-7. Input-output-curve envelope as limited by load factor.

$$\frac{CF}{LF} = \frac{60}{75} = 0.8 = \frac{L_{\max}}{Cap}$$

or the peak load for this condition is 80 per cent of the capacity. Then, finding 80 per cent capacity factor (which is equal to 80 per cent of capacity) on the input-output curve, a straight line is drawn to the minimum load at 10 per cent capacity factor. Where this line intersects with the 60 per cent capacity factor ordinate lies the upper limit of average input rate for a 75 per cent load factor. By finding the limit at all values of capacity factor for 75 per cent load factor this locus can be defined. A similar procedure can find the locus for any selected value of load factor. Several such loci are shown in the figure. The corresponding heat-rate curves are illustrated in Fig. 32-8. By defining the minimum load factor of a load curve the possible range of values of average input rates is appreciably diminished.

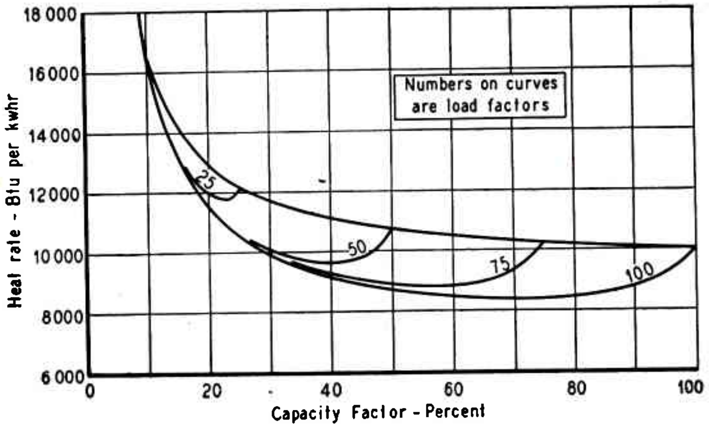


FIG. 32-8. Station heat-rate envelope as limited by load factor.

32-4. Load Division. A system having more than one unit of similar generating equipment has the proper division of load as a problem. Improper load division may appreciably decrease the thermal efficiency of the system as a whole. Figure 32-9 represents the input-output curves of two units that are to operate in parallel and supply a common load. Since A is obviously more efficient throughout its load range, the tendency would be to jump to the incorrect conclusion that A should be loaded first to capacity, then unit B.

The problem is correctly attacked by plotting the *sum* of the inputs to A and B against the load on A for a given constant load on the two units as in Fig. 32-10. For a total load of 8 mw on the two units, in placing zero load on A and 8 mw on B, the total combined input is 148 million Btu per hr. As the load is shifted from B to A, maintaining the combined unit output constant, the total input decreases until A carries 5 mw and B carries 3 mw. Shifting more load onto A then increases the combined input. For maximum economy this load should be divided so as to require the minimum combined input. By plotting a series of constant combined output loci it would be possible to determine the division of load for the entire range of capacity of the two units, as shown in Fig. 32-10.

This method is cumbersome and becomes unwieldy when more than

two units run in parallel. The basic principle of economic load division, demonstrated in the following,* leads to a simple solution for the general

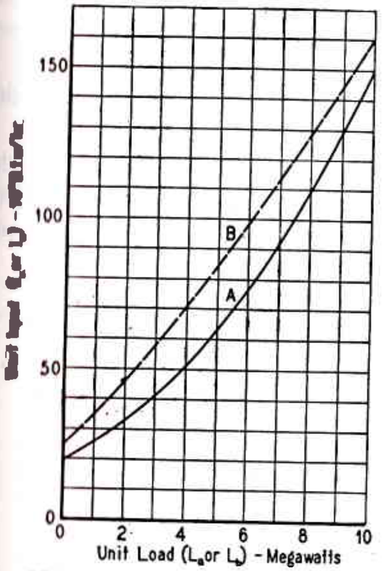


FIG. 32-9. Input-output characteristics of units A and B.

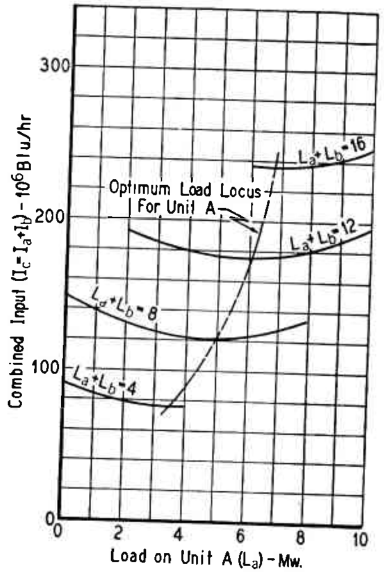


FIG. 32-10. Variation of combined input with varying load division between units A and B of Fig. 32-9.

case: Taking any combined constant-output locus in Fig. 32-10, at the point of minimum input,

$$\frac{dI_c}{dL_a} = 0 \tag{32-11}$$

where

$$I_c = I_a + I_b = \text{combined input to } A \text{ and } B \tag{32-12}$$

$$L_c = L_a + L_b = \text{combined output of } A \text{ and } B \tag{32-13}$$

Then

$$\frac{dI_c}{dL_a} = \frac{dI_a}{dL_a} + \frac{dI_b}{dL_a} = 0 \tag{32-14}$$

$$\frac{dI_a}{dL_a} = -\frac{dI_b}{dL_a} \tag{32-15}$$

But

$$\frac{dI_b}{dL_a} = \frac{dI_b}{dL_b}\frac{dL_b}{dL_a} \tag{32-16}$$

From (32-13)

$$\frac{dL_b}{dL_a} = \frac{dL_c}{dL_a} - \frac{dL_a}{dL_a} \tag{32-17}$$

And since L_c is a constant,

$$\frac{dL_b}{dL_a} = -1 \tag{32-18}$$

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* M. J. Steinberg and T. H. Smith, The Theory of Incremental Rates . . . , *Electrical Engineering*, March, 1934.

Substituting in (32-16),

$$\frac{dI_b}{dL_a} = -\frac{dI_b}{dL_b} \tag{32-19}$$

Substituting in (32-15),

$$\frac{dI_a}{dL_a} = \frac{dI_b}{dL_b} \tag{32-20}$$

or, for minimum combined input to carry a given combined output, the slopes of the input-output curves for each unit must be equal. But the slopes of these curves are the incremental rates which have been previously demonstrated and correlated with the input-output curve.

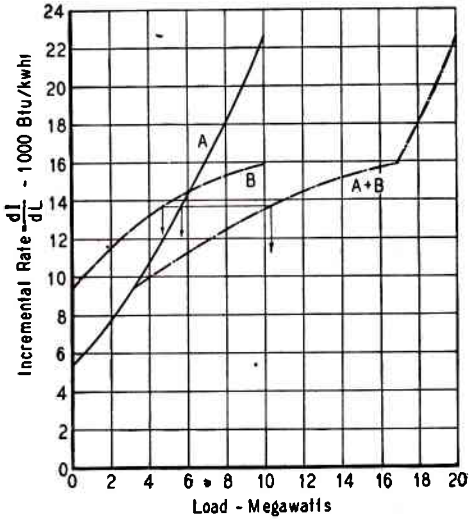

FIG. 32-11. Incremental rates of units *A* and *B* and their combined incremental-rate curve.

Then by plotting the incremental-rate curves as in Fig. 32-11 for the two units *A* and *B* of Fig. 32-9 all that is necessary to divide the load economically is to maintain the incremental rates equal and select the loads accordingly. Thus at an incremental rate of 12,000 Btu per kwhr the load on *A* should be 4,700 kw and on *B* it should be 2,400 kw to generate a combined load of 7,100 kw.

To expedite load division for varying total loads, the combined incremental-rate curve is plotted by adding up the loads at each incremental rate and plotting the sums against the particular incremental rate. The curve *A* + *B* is such a combined curve as in Fig. 32-11. From zero to 3,200 kw total load the combined curve coincides with that for curve *A*. In this range unit *A* takes all the load while *B* operates at zero load.

From 3,400 kw up the total curve progresses upward toward the right to 16,900 kw, at which point *B* reaches full load. The section of the total curve following this load is a reproduction of the upper portion of the *A* curve. To divide the total load of 10,400 kw, the intersection of the load ordinate with *A* + *B* is found; following across at constant incremental rate to the intersections with *A* and *B*, we find that *A* should carry 5,700 kw and *B* should carry the remainder, 4,700 kw.

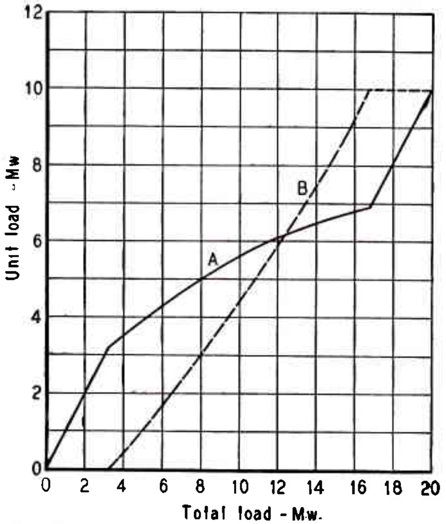

FIG. 32-12. Load-division schedule of units *A* and *B*.

This method of load division can be extended to include any number of units. The loading schedule with the aid of these curves may be presented in tabular form as in Table 32-1 or in graphical form as in Fig. 32-12.

TABLE 32-1. LOAD-DIVISION SCHEDULE, UNITS *A* AND *B*
(Load, megawatts)

Total	Unit *A*	Unit *B*	Total	Unit *A*	Unit *B*
2	2.0	0.0	12	6.1	5.9
4	3.5	0.5	14	6.5	7.5
6	4.3	1.7	16	6.9	9.1
8	5.0	3.0	18	8.0	10.0
10	5.5	4.5	20	10.0	10.0

These input-output curves and derived incremental-rate curves discussed in the foregoing figures are typical of reciprocating steam engines and steam generators and also diesel engines. Multivalved steam turbines have input-output curves like those illustrated in Fig. 32-13. This is a rather complex shape and involves sudden changes of slope at the initiation of the valve openings and points of inflection in the separate sections of the curve beyond the initial valve openings. The incremental-rate curve of the true input-output curve becomes very complex and difficult to use. For most purposes it is sufficiently accurate to simplify the input-output curve by connecting the maximum valve-opening points by straight lines as shown dotted in Fig. 32-13. The relatively slight deviation of this modified characteristic from the true one is of no practical moment. Owing to the constant slopes between valve-opening points, the resulting incremental-rate curve has a step-shaped appearance, as in Fig. 32-14. The vertical sections of the curve merely indicate that the horizontal sections all refer to the same machine; actually the change in slope of the input-output curve at the valve-opening points does not pass through all the intermediate values suggested by the vertical connecting lines.

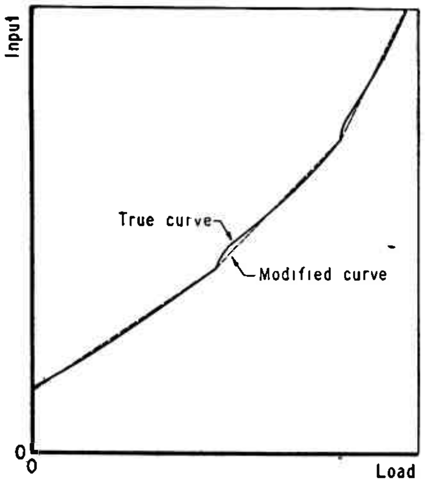

FIG. 32-13. Modification of steam-turbine input-output characteristics.

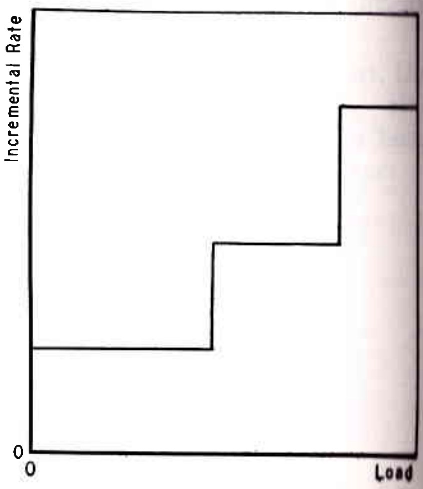

FIG. 32-14. Incremental-rate curve derived from modified steam-turbine characteristics.

In dividing load between turbines of this type to achieve maximum fuel economy, the principle of equal incremental rates still prevails. Since the incremental-rate curves are of a discontinuous nature, it is generally impossible to hold the turbine increment rates at equal values. Fundamentally the principle of equal incremental rates may also be regarded as apportioning an increase in load to the machine or machines that will incur the minimum increase in input. For continuous incremental-rate curves this results in maintaining the incremental rates of all machines equal at all loads. For discontinuous incremental-rate curves at any total load the machine with the lowest incremental rate picks up the load. Figure 32-15 shows the incremental rates for two turbines A and B. Starting at zero combined load, machine B picks up to load L_1 with A remaining at zero load for a total load of L_1. With increasing total load, B remains at L_1 and A picks up load to L_2 for a total load of $L_1 + L_2$. The combined incremental-rate curve of the two machines is the dot-dash line. For any increase above $L_1 + L_2$, A carries it on its second valve to L_3 for a total load of $L_1 + L_3$. Note that the combined incremental-rate curve for the two machines is of an increasing nature, indicating that at all loads the increase is picked up at the maximum possible economy.

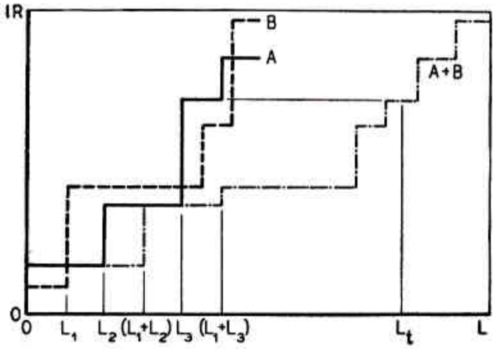

FIG. 32-15. Example of discontinuous incremental-rate curves and their combined curve.

To divide load L_t between two machines, we find that it intersects the combined curve at the sixth step. Reading to the left, we find that the variation in load is being taken on the third valve of A and that B is carrying load at the maximum point for its third valve. For any number of steam turbines, if none of the steps have the same incremental rate, the combined incremental-rate curve will contain the same number of steps as the sum of the steps in the individual incremental-rate curves.

32-5. Station Performance Characteristic. The performance of any individual piece of power-plant equipment such as a boiler, turbine, diesel engine, pump, fan, or heat exchanger may be described by an input-output curve. Since a station is made up of several of these different forms of apparatus, their performance characteristics must be integrated so that the performance of the station as a whole may be expressed by a single input-output curve.

Figure 32-16 shows the major curves for a steam station. A boiler curve assumes the general form heretofore considered, but both ordinates will be scaled in Btu per hour. The input will be the product of the fuel-flow rate, and its calorific value, and the output will be the product of the steam or water flow and its enthalpy rise through the boiler. In this case two boilers, or steam generators, are feeding a turbine. The curve for one boiler and for both boilers combined is indicated on the graph. Since both boilers are identical, the combined boiler curve is easily reproduced by merely doubling all values for the single boiler curve.

The turbine-generator curve has a somewhat more complex form, depending upon the number of valves and their arrangement. The curve is for a unit having three sets of valves. The reason for the trends of this curve is discussed in Chap. 12. The input is in Btu per hour and the output in kilowatts.

The energy required to drive the auxiliary equipment of the station varies with the station load. The correlation is often of the simple $y = a + bx$ form, and the slope may be relatively flat. The auxiliary energy requirements for the entire station are plotted against the turbine-generator or gross station load (Fig. 32-16*c*). This curve may be readily converted to the form of station net load plotted against gross load as indicated in Fig. 32-16*d*.

The method of combining these curves is based on tracing a given magnitude of energy entering the boilers (and therefore the station) through the station equipment and finding the amount that finally emerges as useful energy for transmission and distribution. Starting in Fig. 32-16*a* with a block of energy I_1 as input to the boilers, a certain portion O_1 appears as output leaving in the steam which in turn becomes the turbine input in curve *b*. This input O_1 produces a block of energy at the generator terminals, L_1 kw. Some of this energy must be diverted for auxiliary operation, the remainder being available as the final product of the station. Going to curve *d* with L_1 gross kilowatts, find L_2 net kilowatts available for distribution. Then, by plotting L_2 kw against I_1 Btu per hr input in curve *e*, the performance characteristic of the station as an integral unit is determined.

32-6. Station Incremental Rate. The incremental rate of the station as a unit may be found directly from the resulting input-output curve as in Fig. 32-16*e*. It is often more convenient and somewhat more accurate to calculate the station incremental rate from the individual equipment incremental rates.*

Let

dI/dL_n = station incremental rate = R_{st}
dI/dO = steam-generator incremental rate = R_b
dO/dL_g = turbogenerator incremental rate = R_t
dA/dL_g = auxiliary consumption incremental rate = R_a

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* M. J. Steinberg and T. H. Smith, The Theory of Incremental Rates . . . , *Electrical Engineering*, April, 1934.

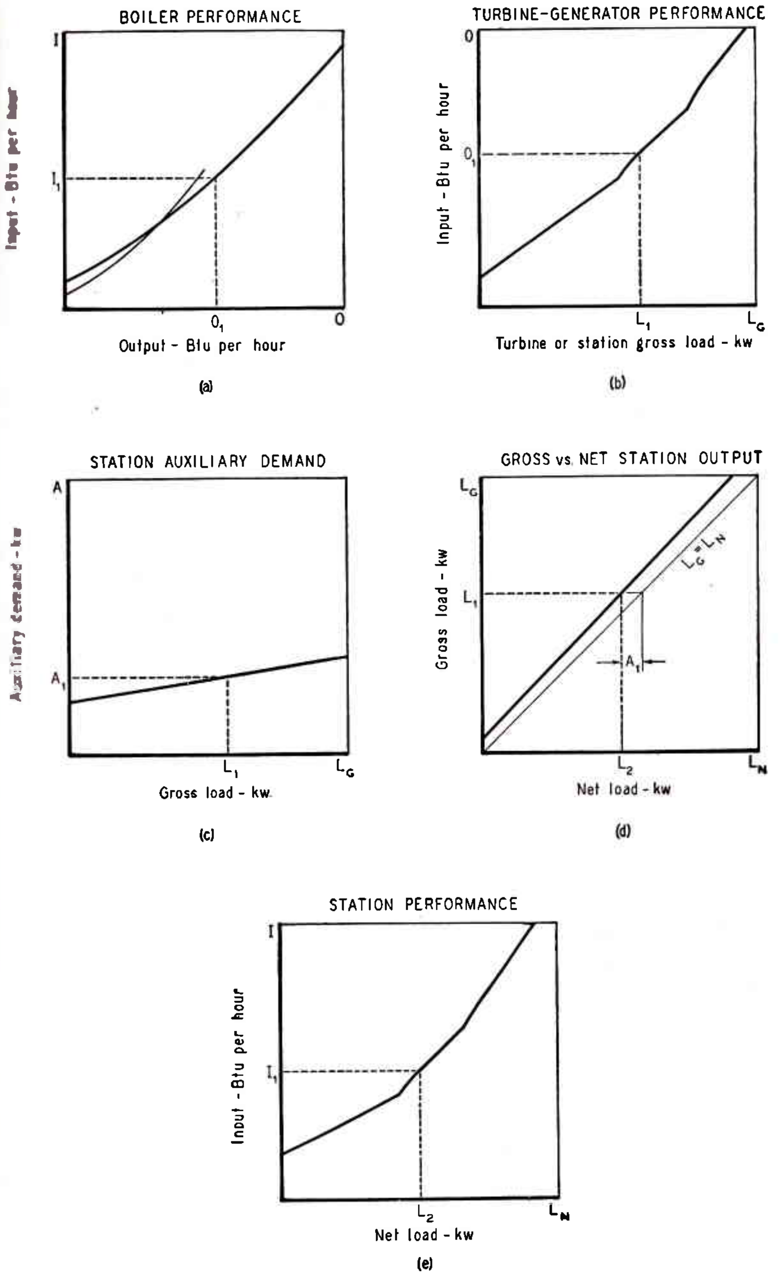

FIG. 32-16. Input-output curves of component station equipment and derivation of station input-output curve.

Then, by identity,

$$\frac{dI}{dL_n} = \frac{dI}{dO} \times \frac{dO}{dL_g} \times \frac{dL_g}{dL_n}$$

$$R_{st} = R_b \times R_t \times \frac{dL_g}{d(L_g - A)}$$

$$= R_b \times R_t \times \frac{1}{d(L_g - A)/dL_g}$$

$$= R_b \times R_t \times \frac{1}{1 - R_a} = \frac{R_b R_t}{1 - R_a}$$

By following the incremental-rate curves of the individual equipment in a manner similar to that for the-input-output curves of Fig. 32-16 the station incremental rate can be computed and plotted.

32-7. Capacity Scheduling. In a power plant with several units most of the units need be in operation only when the peak demand occurs. A schedule is needed to indicate the order and the total plant or system load at which the various units should be placed in operation. To establish a basis for reasonable continuity of service, sufficient capacity should be operating at all times so that in case of failure of the largest unit of capacity in service the remaining active units should be capable of maintaining energy supply without interruption.

If the minimum load is equal to or less than the capacity of a single unit, two units will have to be placed in operation to protect continuity of service. The order in which the units are placed in service depends upon their relative efficiencies. The most efficient machine is placed in service first, followed by machines in descending order of efficiency as needed. The load is divided among the units in operation on the basis of equal incremental rates.

As an example, a system has the units of capacity shown in the following tabulation, with the indicated order of efficiency:

Unit No.	Capacity, mw	Order of efficiency
1	5	7
2	10	5
3	10	6
4	20	4
5	50	2
6	50	1
7	70	3

For the given conditions the capacity schedule would appear as in the following tabulation:

CAPACITY IN OPERATION

Mw system load	Unit							Total
	1	2	3	4	5	6	7	
0	. .	. .	. .	. .	50	50	. .	100
50	. .	. .	. .	. .	50	50	. .	100
50+	. .	. .	. .	. .	50	50	70	170
100	. .	. .	. .	. .	50	50	70	170
100+	. .	. .	. .	20	50	50	70	190
120	. .	. .	. .	20	50	50	70	190
120+	. .	10	. .	20	50	50	70	200
130	. .	10	. .	20	50	50	70	200
130+	. .	10	10	20	50	50	70	210
140	. .	10	10	20	50	50	70	210
140+	5	10	10	20	50	50	70	215

From loads of 0 to 50 mw two units run; upon the loss of one the remaining unit could carry the load in this range. From loads in excess of 50 mw the loss of the largest unit, 70 mw, would not shut down the system because of lack of active capacity. Note that, though the capacity of the system is 215 mw, the maximum *protected* demand that it can carry is 215 − 70 = 145 mw. The manner in which the capacity would vary over a day is illustrated in Fig. 32-17.

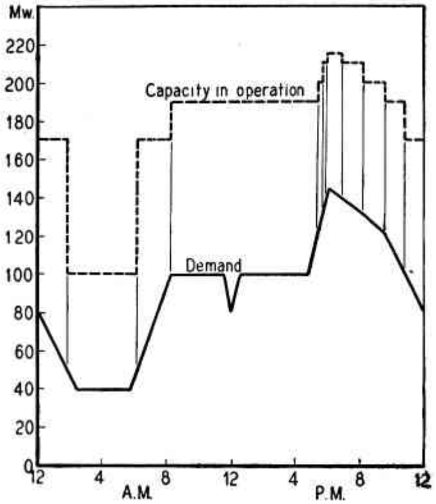

FIG. 32-17. Hourly variation of capacity with demand.

The turbine or engine capacity in active operation is termed the "spinning" capacity. The spinning capacity for minimum protection, as shown in the preceding, should always be greater than the load. The difference between spinning capacity and the load is termed the "spinning reserve." The minimum magnitude of the spinning reserve is generally equal to the capacity of the largest unit. This reserve does not designate certain specific units of the group but is merely a portion of the total spinning capacity that protects against the accidental loss of the largest unit.

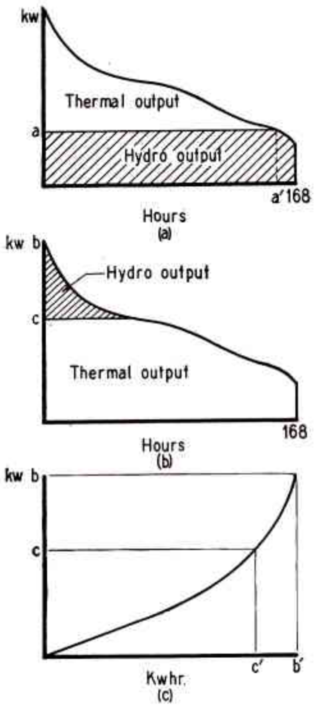

FIG. 32-18. (*a*) Base-load operation of hydraulic plant during maximum river flow; (*b*) peak-load operation of hydraulic plant during minimum river flow; (*c*) energy-load curve for (*a*) and (*b*).

In some power systems having a relatively large number of units the total installed capacity as a minimum is not allowed to be less than the sum of the peak demand expected plus the capacity of the two largest units; i.e., the installed reserve is maintained equal to the capacity of the two largest units. There is no general fixed rule regarding the required installed reserve; some systems specify three units, and others a certain percentage of the expected peak demand, while others depend on a probability analysis (Sec. 31-2).

32-8. Hydraulic-station Operation. The question of load division and capacity scheduling for a pure hydraulic system not using any thermal equipment is subject to the water flow available and the peak demand to be met. In a properly designed hydraulic system sufficient water flow is normally available to meet the demand at any time. At times of low flow it may require careful scheduling of units to maintain optimum efficiency at all load levels to meet the system demand.

When a hydraulic station is one unit of a system containing both hydraulic and thermal stations, its operation will generally be handled in a somewhat different fashion. Since the fuel-cost component is absent in hydraulic operation, the water-generated energy would be used whenever possible. When the river has ample flow, the plant is loaded continuously to produce the maximum energy possible with its equipment to carry the *base of the load curve* (Fig. 32-18*a*). The plant capacity or the energy in the river flow, whichever is the smaller, is *a* kw. Whenever the load drops below the plant capacity or the energy in the river flow, as beyond *a*′ hr, some of the water may flow over the dam spillway if storage is not available.

At periods of low flow careful attention should be given to the available water flow and its use so that the hydraulic-plant capacity is available to help carry the peak load of the system. This is usually done by shutting down the station and storing the river flow during off-peak hours as pondage and then at the time of the peak period placing the station in operation by drawing upon the pondage. It is generally possible to estimate closely the amount of river flow available for a coming week and, therefore, the amount of energy. The system load at which the hydraulic station should be cut into operation is estimated in the following manner: With the probable load curve for the coming week known as in Fig. 32-18*b*, the load-energy curve as in Fig. 32-18*c* can be calculated. If the amount of energy in the river flow is equal to $b' - c'$ kwhr, this amount of energy is supplied to the load curve in its upper load range. Then, upon laying off $b' - c'$ kwhr from the total energy in the load curve on the load-energy curve in *c*, it is found that this amount of energy is contained in the area above *c* kw. Thus, whenever the load exceeds *c* kw, the hydraulic station is cut into operation and carries the variable portion of the system load while the thermal plants carry a constant load. The difference $b - c$ kw must not exceed the capacity of the hydraulic station. If it does, this means that the thermal plants will increase their load at the time of the peak load to prevent overloading the hydraulic station; the hydraulic station will be cut in at a lower load than for the former case.

PROBLEMS

32-1. An input-output curve of a 100-mw station is expressed by the formula

$$I = 10^6(180 + 10L + 0.001L^3)$$

where I is in Btu per hour and L is in megawatts.

a. Find the load at which the maximum efficiency occurs without plotting any curves.

b. Plot the heat-rate and thermal-efficiency curves.

32-2. For the station in Prob. 32-1 compute the input from 0 to 100 mw in 10-mw steps.

a. Assuming that the equation of the input-output curve is unknown, compute and plot the incremental-rate curve for the station from these data.

b. Compute and plot on the same graph the true incremental-rate curve.

32-3. The input-output curve of a 150-mw station is given by

$$I = 10^6 \times (250 + 5L + 0.03L^2) + L^4$$

a. Plot the input-output, heat-rate, and incremental-rate curves.

b. Calculate the load at which minimum heat rate occurs, and check with the plot.

c. What is the area under the incremental-rate curve in Btu per hour between the loads 0 and 120 mw?

d. What is the additional input to generate 20 mw above 120 mw?

32-4. A 10-mw diesel engine has the following heat-rate curve,

$$HR = 10^6 \times \left(\frac{20}{L} + 4 + 0.3L + 0.01L^2\right)$$

where HR is in Btu per megawatthour and L is in megawatts. The annual load-duration curve is linear between the given points of Table P-1 for the above engine.

Table P-1

Load, mw	Hours duration	Load, mw	Hours duration
10.0	0	5.0	4,500
8.0	500	5.0	5,500
7.6	1,000	2.0	7,000
7.4	3,000	1.0	8,000
7.0	3,500	0.0	8,000

At zero load the engine is shut down.

a. Find the annual average heat rate for the engine, using the average loads for 2,000-hr intervals.

b. Find the average heat rate, using 1,000-hr intervals.

c. Plot the heat-rate envelope for the engine between loads of 1.0 and 10.0 mw.

d. Plot the average heat rate from *b* in the envelope.

32-5. Turbogenerator units *A* and *B*, each of 10 mw capacity, have the following input-output characteristics,

Unit *A*: $I_a = 10^6 \times (10 + 9L_a)$
Unit *B*: $I_b = 10^6 \times (20 + 9L_b)$

where I is in Btu per hour and L is in megawatts.

a. If either unit were to be placed in operation alone which should be selected? Why?

b. Devise a loading schedule in 5-mw steps for the most economical operation when both units are operating in parallel.

32-6. A system has three generating units with the curves given in Table P-2.

Table P-2

Unit	Capacity, mw	Input-output curve
A	10	$I = 10^6 \times (13 + 0.04L^2 + 0.02L^3 + 9L)$
B	10	$I = 10^6 \times (10 + 8L + 0.4L^2)$
C	8	$I = 10^6 \times (16 + 8L)$

a. Plot the input-output curves of all units on a common graph.

b. Plot the heat-rate curves of all units on a common graph.

c. Plot the individual and combined incremental rates on a common graph.

d. Devise a combined loading and capacity schedule for the three units from 0 to 28 mw total load, maintaining a one-unit reserve where possible.

32-7. A single-unit station has one steam generator supplying a 30-mw turbogenerator with all auxiliaries driven electrically. For practical purposes the turbine input-output curve of Table P-3 may be considered linear between zero load and full valve openings.

Table P-3

Valve opening	Load, mw	Input, 10^6 Btu per hr
Partial primary	0	30
Maximum primary	15	135
Maximum secondary	20	185
Maximum tertiary	30	300

The steam-generator input-output curve is

$$I = 15 + 1.05L + \frac{L^3}{900{,}000}$$

where I and L are in millions of Btu per hour.

With the turbine operating at zero load the station auxiliaries require 500 kw; at full load they require 1,500 kw.

a. Calculate the station inputs at turbine gross megawatt loads of 0.5, 5, 10, 15, 17.5, 20, 25, and 30.

b. From a plot of the computations in (*a*) determine the station incremental-rate curve.

c. From the individual equipment increment rates compute the station increment rate, and compare with (*b*) on a common graph.

d. Compute and plot the station heat-rate curve from (*a*).

32-8. An electrical supply system has a 15-mw diesel station and a 5-mw hydraulic plant as sources of energy. The hydraulic plant has pondage provisions and can store sufficient water to generate 72 mwhr. For the coming week the estimated power in the river flow is 1.5 mw; the expected system loads are as in Table P-4.

Table P-4

Load, mw	*Hr at load*
20	5
15	70
12	50
8	20
5	23

a. Find the total energy that the hydraulic plant can generate.

b. Plot the system load-duration curve, and show the division of load between the hydraulic and the diesel station.

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CHAPTER 33

SPECIFIC ECONOMIC ENERGY PROBLEMS

33-1. Introduction. In the foregoing chapters we discussed the fundamentals of engineering economics and the factors to be considered in making a selection of alternative methods of power production. This chapter presents specific problems in the power-plant field. This discussion is not exhaustive but is intended to demonstrate the technique of economic energy studies.

Most of the problems treated herein have been stripped to their barest fundamentals to demonstrate the application of basic principles. In practice the solution of these problems follows the elementary patterns outlined, but very often the factors that are "given" in these discussions actually require a great deal of time to determine. Indeed, to collect these facts properly and where necessary make assumptions and predictions that are acceptable are tests of an engineer's ability.

33-2. Selection of Steam Plant. When circumstances determine that a given type of plant is to be selected, such as a steam plant, the comparisons that must be made are reduced in number. The variables existing within one type of plant are generally less than the number of variables among different types of plant.

Example 33-1. An industrial factory requires a source of energy to meet a 10,000-kw maximum demand. The annual duration of the load is estimated as in the following tabulation:

Load, kw	*Hr at load*
10,000	1,000
7,000	4,000
4,000	2,000
1,000	1,000
0	760

As service interruptions can be tolerated, no spare capacity need be provided. Coal is available at $8 per short ton and has a calorific value of 13,000 Btu per lb. Since oil is available only at prohibitive prices, diesel engines are not considered. After preliminary studies, four different steam plants are being considered as shown in Table 33-1.

The annual operating costs of labor, supervision, supplies, repairs, and operating taxes are expected to be the same for all proposals. If the fixed charge rate is 15 per cent, determine the economic plant.

TABLE 33-1. COMPARATIVE COSTS AND PERFORMANCE OF STEAM STATIONS

Plant	Steam conditions	Capacity cost per kw[a]	Input-output characteristics
A	600 psi 750 F	$136	$I = 10^6 \times (13 + 9L + 0.04L^2 + 0.02L^3)$
B	800 psi 850 F	$140	$I = 10^6 \times (11 + 8.5L + 0.03L^2 + 0.015L^3)$
C	1,200 psi 900 F	$145	$I = 10^6 \times (10.2 + 8.2L + 0.025L^2 + 0.012L^3)$
D	2,400 psi 950 F	$160	$I = 10^6 \times (10 + 8L + 0.02L^2 + 0.01L^3)$

I is in Btu per hour and *L* is in megawatts.

[a] Excluding land, structures, and electrical system.

Solution

$$\text{Unit fuel cost} = \frac{800 \times 1,000,000}{2,000 \times 13,000} = 30.76 \text{ cents per million Btu}$$

The first step is to determine the total energy input requirements for each plant from the given input-output characteristic and the load data as in Table 33-2.

TABLE 33-2. CALCULATION OF COMPARATIVE FUEL REQUIREMENTS OF PROPOSED PLANTS

Load-energy data			Plant *A*		Plant *B*		Plant *C*		Plant *D*	
Load, mw	Hr	Mwhr	Input rate, 10^6 Btu per hr	Total input, 10^9 Btu	Input rate, 10^6 Btu per hr	Total input, 10^9 Btu	Input rate, 10^6 Btu per hr	Total input, 10^9 Btu	Input rate, 10^6 Btu per hr	Total input, 10^9 Btu
10	1,000	10,000	127.00	127.00	114.00	114.00	106.70	106.70	102.00	102.00
7	4,000	28,000	84.82	339.28	77.12	308.48	72.95	291.80	70.41	281.64
4	2,000	8,000	50.92	101.84	46.44	92.88	44.17	88.34	42.96	85.92
1	1,000	1,000	22.06	22.06	19.54	19.54	18.44	18.44	18.03	18.03
Totals		47,000	......	590.18	......	534.90	......	505.28	......	487.59
Heat rate, Btu per kwhr			......	12,560	......	11,380	......	10,750	......	10,370
Fuel cost			......	$181,600	......	$164,600	......	$155,400	......	$150,000

The megawatthours are the product of the load and the number of hours the load persists. The input rate is calculated from the input-output characteristic for the given megawatt load. The total input at each load is the product of the input rate and the number of hours at the given load. As a summary of the load conditions note that the load factor is

$$LF = \frac{47,000,000}{8,760 \times 10,000} = 0.537, \text{ or } 53.7\%$$

Since the fuel requirement is the only factor of the operating cost that varies from plant to plant, it is the only component that need be considered in the relative annual costs. Then

$$C_o = \text{fuel input} \times 10^{-6} \times 0.3076$$
$$C_f = \text{unit capacity cost} \times 10,000 \times 0.15$$
$$C = C_o + C_f$$

The relative annual costs of the alternate plants are shown in Table 33-3.

TABLE 33-3. ANNUAL COST COMPARISON OF PROPOSED PLANTS

Plant	Total investment	C_f	C_o	C
A	$1,360,000	$204,000	$181,600	$385,600
B	1,400,000	210,000	164,600	374,600
C	1,450,000	217,500	155,400	372,900
D	1,600,000	240,000	150,000	390,000

Plant *C* shows the lowest relative cost and, therefore, is the optimum for the given load conditions, fuel cost, and fixed-charge rate.

With the given information the same conclusion can be shown by two other methods: rate of return available for fixed charges on increment investment and justifiable added investment. These methods are illustrated in Tables 33-4 and 33-5, respectively. In Table 33-4 the

TABLE 33-4. RATE OF RETURN AVAILABLE FOR FIXED CHARGES ON INCREMENT INVESTMENT

Plant	Investment	ΔP	C_o	ΔC_o	$R = \Delta C_o/\Delta P$
A	$1,360,000		$181,600		
		$ 40,000		$17,000	0.425
B	1,400,000		164,600		
		50,000		9,200	0.184
C	1,450,000		155,400		
		150,000		5,400	0.036
D	1,600,000		150,000		

rates *R* are comparable with the given rate of 15 per cent; hence the rate of return on increment investment from *B* to *C* is obviously profitable; for any increment above plan *C* the available rate of return is considerably below the desired value. Hence plan *C* is the optimum.

In Table 33-5 the columns for ΔC_o and actual ΔP are copied from Table 33-4. A comparison of the justifiable added investments ΔP with the actual added investments shows that *C* costs only $50,000 additional against a permissible $61,300 and is obviously a good investment; but

TABLE 33-5. JUSTIFIABLE ADDED INVESTMENT

Plant	ΔC_o	$\Delta P = \Delta C_o/R$	Actual ΔP
A			
	$17,000	$113,200	$ 40,000
B			
	9,200	61,300	50,000
C			
	5,400	36,000	150,000
D			

again any additional investment above *C* is much greater than the permissible justifiable investment. Hence *C* is the optimum. This method of comparison is sometimes used by organizations that wish to avoid any possible collusion between engineering groups evaluating the bids for the purchaser and the manufacturers offering bids for furnishing equipment. In this type of arrangement the purchaser furnishes the evaluation engineers all the necessary data received from the manufacturers *except* the actual purchase price. The engineers then evaluate the justifiable added investments of all bids relative to each other and submit their conclusions to the purchaser. Arranging the bids in ascending order of investments or purchase price, the purchaser then compares the actual differences with relative justifiable added investments submitted by the engineers and makes his own selection.

In this type of problem the methods of rate of return, rate of return available for fixed charges, and justifiable investment are not applicable. These three methods are suitable only where a choice exists between (1) buying a service from someone and (2) making an investment in order to supply that service for oneself. In the problem discussed it was implied that the required energy (or service) could not be purchased but that it was necessary to invest in station equipment to supply the needed energy. In other words, an investment had to be made, and no possibility existed of paying an annual charge in lieu of making such an investment.

33-3. Selection from Different Types of Prime Movers. When the types of plant differ, various items of the total annual cost will differ in the various schemes. In Sec. 33-2, Selection of Steam Plant, the principal differing costs were the initial investment and the fuel input. However, when a diesel plant is compared with a steam plant in addition to these items, the diesel plant often requires less labor; also, the unit fuel prices differ. For correct analysis these differing costs must all be included in the relative annual costs.

Example 33-2. To serve a load having the annual duration characteristics given in the following tabulation:

Kw	*Hr*
5,000	200
4,000	4,000
2,000	2,000
1,000	1,000
500	1,560

Two plants, a steam-turbine plant and a diesel-engine plant, are being considered. For the steam plant coal at $4.50 per short ton with a calorific value of 14,000 Btu per lb is available; for the diesel plant oil at $2.10 per barrel containing 358 lb with a calorific content of 18,500 Btu per lb.

The performance characteristics of the plants are

Steam: $I = 10^6 \times (6 + 8L + 0.111L^3)$
Diesel: $I = 10^6 \times (9 + 4L + 0.52L^2 - 0.016L^3)$

The steam plant would require three more men at a salary of $4,000 per man for operation than the diesel plant. The fixed charge rate is 12 per cent for each plant. Which plant should be selected if the steam plant costs $230 per kilowatt and the diesel plant $220 per kilowatt, and no reserve capacity is required?

Solution

$$\text{Coal unit fuel cost} = \frac{450 \times 1{,}000{,}000}{2{,}000 \times 14{,}000} = 16.07 \text{ cents per million Btu}$$

$$\text{Oil unit fuel cost} = \frac{210 \times 1{,}000{,}000}{358 \times 18{,}500} = 31.71 \text{ cents per million Btu}$$

By summarizing the varying relative costs with the aid of Table 33-6 the optimum plant can be determined.

TABLE 33-6. CALCULATION OF FUEL REQUIREMENTS

Mw	Hours	Mwhr	Steam plant		Diesel plant	
			Input rate, 10^6 Btu per hr	Total input, 10^9 Btu	Input rate, 10^6 Btu per hr	Total input, 10^9 Btu
5	200	1,000	59.90	11.98	40.00	8.00
4	4,000	16,000	45.12	180.48	32.30	129.20
2	2,000	4,000	22.89	45.78	18.95	37.90
1	1,000	1,000	14.11	14.11	13.50	13.50
½	1,560	780	10.01	15.62	11.13	17.36
Total. .	8,760	22,780	. . . . .	267.97	. . . . .	205.96
Average heat rate, Btu per kwhr.			. . . . .	11,800	. . . . .	9050

Steam plant:

$$C_f = PR = 230 \times 5{,}000 \times 0.12 = \$138{,}000$$
$$C_o = (267{,}970 \times 0.1607) + (3 \times 4{,}000) = \$55{,}060$$
$$C = 138{,}000 + 55{,}060 = \$193{,}060 \text{ annually}$$

Diesel plant:

$$C_f = PR = 5{,}000 \times 220 \times 0.12 = \$132{,}000$$
$$C_o = 205{,}960 \times 0.3171 = \$65{,}310$$
$$C = 132{,}000 + 65{,}310 = \$197{,}310 \text{ annually}$$

Note that the annual costs are only relative since they compare only the differential costs. All the costs of equal magnitude in each scheme are not included in the figures.

The steam plant would be the choice under the given circumstances despite the higher investment and the greater relative labor cost. These items are outweighed by the cheaper fuel cost.

Since fuel prices vary from time to time, the price of oil more so than that of coal, comparisons depend upon the relative fuel prices at the time the study is made. To give a complete picture of the weight of the fuel prices in determining plant desirability, the cost comparison can be set up to determine the relationship of the fuel prices for given conditions of labor costs, thermal efficiency, and load conditions.

From the foregoing conditions, it is evident that the plants are equally desirable when their annual costs are equal.

$$C_{\text{steam}} = C_{\text{diesel}}$$
$$(PR + C_o)_{\text{steam}} = (PR + C_o)_{\text{diesel}}$$
$$(1{,}150{,}000 \times 0.12) + \frac{267{,}970 \times 10^6 \times S}{2{,}000 \times 14{,}000} + 12{,}000$$
$$= (1{,}100{,}000 \times 0.12) + \frac{205{,}960 \times 10^6 \times D}{358 \times 18{,}500}$$

where S = price of coal, dollars per ton
D = price of oil, dollars per bbl

Solving,

$$D = 0.578 + 0.308S$$

Then, for coal at \$4.50 per ton, oil would have to be \$1.39 per barrel for parity. To sway the decision toward a diesel plant the oil price of course should be somewhat less than this value. Conversely, for oil at \$2.10 per barrel, coal would be \$4.94 per ton to establish parity.

33-4. Selection of Unit Sizes.

In providing the units for a station the selection of the unit sizes ought not to be a matter of whim or chance but should be the result of a careful economic study. The unit sizes depend upon the character of the load as well as on the unit costs and thermal economy.

Example 33-3. It has been decided that a diesel plant is the best type for a certain installation. The maximum demand to be met is 9,000 hp. The available sizes of diesel units are 10,000; 5,000; 3,000; 2,000; and 1,000 hp. The load is one requiring a moderate degree of protection against accidental unit outage, and it has been decided that a one-unit reserve will be adequate.

The unit capacity costs for the various sizes are as shown in the following tabulation:

Hp	*Cost per hp*
10,000	\$100
5,000	110
3,000	125
2,000	140
1,000	170

The total labor and supplies are assumed to remain constant regardless of the number of units; maintenance is assumed to cost \$800 per unit annually regardless of size. The investment in building and accessory equipment is assumed constant for all cases. Fixed charges are 14 per cent. Fuel costs \$1.50 per barrel of 360 lb and has a heating value 18,000 Btu per lb.

The annual load curve consists of the values given in the following tabulation:

Hp	*Hr*
9,000	100
8,000	200
6,000	3,000
4,000	2,000
2,000	1,000
1,000	2,460

The heat-rate characteristics for the various size of units are plotted in Fig. 33-1*a*. Most diesel engines are most efficient at their full-load points. The heat-rate curves were derived from the following input-output characteristics, where L is the load in thousands of horsepower and I is the input in Btu per hour.

10,000-hp unit: $I = 10^6 \times (14 + 3.8L + 0.29L^2 - 0.007L^3)$
5,000-hp unit: $I = 10^6 \times (9 + 4L + 0.52L^2 - 0.016L^3)$
3,000-hp unit: $I = 10^6 \times (6.5 + 4.5L + 0.5L^2 + 0.037L^3)$
2,000-hp unit: $I = 10^6 \times (4.5 + 5L + 0.5L^2 + 0.19L^3)$
1,000-hp unit: $I = 10^6 \times (2.4 + 5L + 1.8L^2 + 0.3L^3)$

Solution

$$\text{Unit fuel cost} = \frac{150}{360 \times 18{,}000} \times 10^6 = 23.15 \text{ cents per million Btu}$$

From the heat-rate curves of the individual units we see that, for maximum economy, the machines should be kept loaded as near their full ratings as possible. However, the number of machines kept in operation will be dictated by the magnitude of the load and the one-unit reserve requirement. Since the machines have identical characteristics, they should always be kept equally loaded to maintain optimum economy. Since the maximum demand is 9,000 hp, the following number of each size of machine must be installed to provide the required protection: two 10,000-hp; three 5,000-hp; four 3,000-hp; six 2,000-hp; and ten 1,000-hp. The combined heat rates for each of these unit combinations, maintaining a one-unit reserve, are plotted in Fig. 33-1*b*. In the lower range of loads the combination of small units is more efficient; at the higher range of loads the large units are more efficient; in an intermediate range of loads there is not much difference in thermal economy. A load curve with a high load factor and the given peak value will show better thermal economy with the large units, and one with the same peak load and a low load factor will show better thermal economy with the small units.

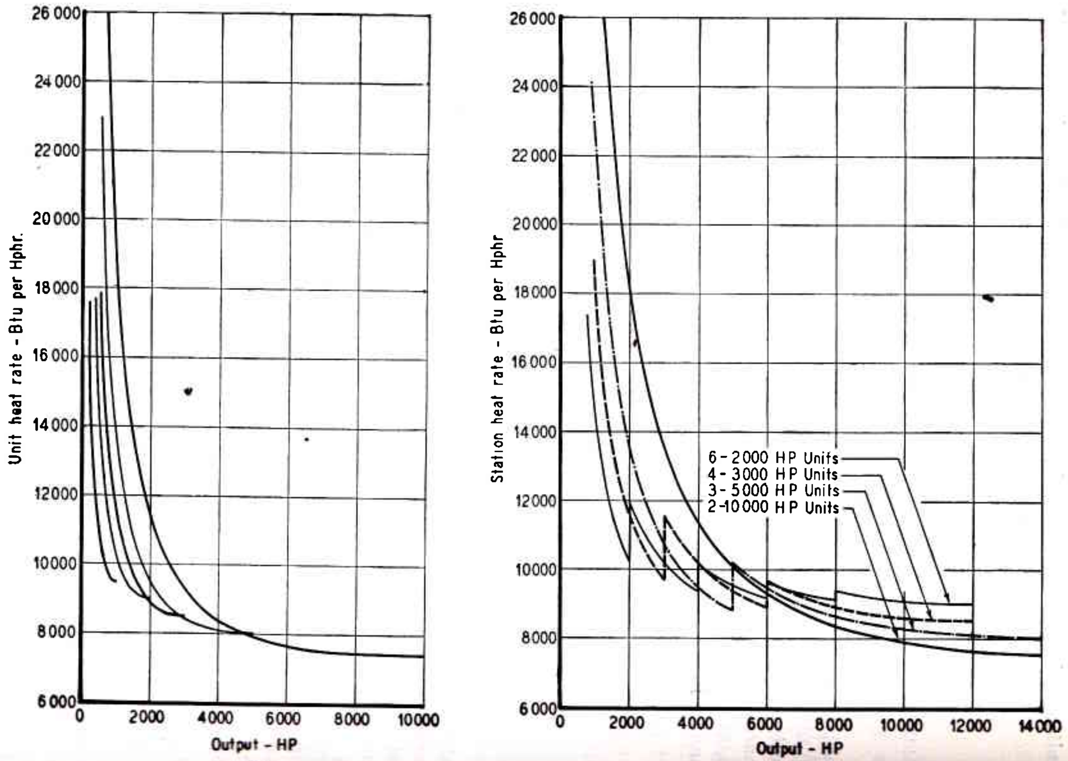

FIG. 33-1. (a) Comparison of diesel-unit heat rates (hypothetical); (b) comparison of diesel-station heat rates.

TABLE 33-7. COMPARATIVE DIESEL-STATION FUEL REQUIREMENTS

L	Hours	Energy	Two 10,000-hp units			Three 5,000-hp units			Four 3,000-hp units			Six 2,000-hp units			Ten 1,000-hp units		
			N	*I*	Σ*I*	*N*	*I*	Σ*I*	*N*	*I*	Σ*I*	*N*	*I*	Σ*I*	*N*	*I*	Σ*I*
9	100	900	2	36.33	7.27	3	25.25	7.58	4	19.56	7.82	6	13.77	8.26	10	8.58	8.58
8	200	1,600	2	33.39	13.36	3	23.00	13.80	4	17.80	14.24	5	14.56	14.56	10	8.48	15.26
6	3,000	18,000	2	27.82	166.92	3	18.95	170.55	3	17.80	160.20	4	13.77	165.24	7	8.20	172.20
4	2,000	8,000	2	22.70	90.80	2	18.95	75.80	3	13.48	80.88	3	12.50	75.00	5	7.71	77.10
2	1,000	2,000	2	18.08	36.16	2	13.50	27.00	2	11.54	23.08	2	10.19	20.38	3	6.62	19.86
1	2,460	2,460	2	15.97	78.57	2	11.13	54.76	2	8.88	43.69	2	7.15	35.18	2	5.39	26.52
Total.....	8,760	32,960	..	.....	393.08	..	.....	349.49	..	.....	329.91	..	.....	318.62	..	.....	319.52
Average heat rate, Btu per hphr					11,930			10,600			10,020			9,670			9,700
Annual fuel cost					$91,000			$80,910			$76,370			$73,760			$73,970

L = load in 1,000 hp.
Hours = hours at load.
Energy = total generation in 10^3 hp-hr at load for given hours.
N = number of units operating.
I = millions of Btu per hour input per unit.
ΣI = billions of Btu total input to unit combination.

The fuel cost for each combination of units is calculated in Table 33-7, in which the number of units required for each load is indicated. The relative annual total costs are calculated in Table 33-8.

TABLE 33-8. COMPARISON OF RELATIVE TOTAL ANNUAL COSTS

Units, hp	No.	Investment	C_f, \$1,000	Fuel cost, \$1,000	Repair cost, \$1,000	Total C, \$1,000	Plant capacity, hp	Per cent capacity factor
10,000	2	\$2,000,000	280.0	91.0	1.6	372.6	20,000	18.8
5,000	3	1,650,000	231.0	80.9	2.4	311.9	15,000	25.1
3,000	4	1,500,000	210.0	76.4	3.2	289.6	12,000	31.4
2,000	6	1,680,000	235.2	73.8	4.8	313.8	12,000	31.4
1,000	10	1,700,000	238.0	74.0	8.0	320.0	10,000	37.6

The plant with four 3,000-hp units proves the most economical. It so happens that it is also the cheapest in point of investment but not the most economical in fuel for the given load curve. The 10,000-hp station, being the smallest, shows the highest capacity factor, but it is unattractive economically despite this fact. The annual load factor for the given curve is

$$LF = \frac{32{,}960{,}000}{9{,}000 \times 8{,}760} = 0.418, \text{ or } 41.8\%$$

Analysis of the components of the relative total annual costs shows that the variation in unit price of the units is the biggest factor in determining the optimum unit size. The resulting effect on the fixed cost shows a difference of \$21,000 annually with the next most economical size of three 5,000-kw units. However, the difference in relative operating costs is only \$3,700 for the same two combinations. It is very unlikely that a different set of load-duration conditions (with the same peak load) will change the conclusion in this particular case by outweighing the \$21,000 fixed-cost differential.

33-5. Capacity Requirement with Respect to Load Growth. The general experience of utility power systems has been one of continuous growth, though at rates varying from year to year. At times, some have suffered temporary declines in peak demand, as during the depression years of 1930 to 1933. In cases of more or less steady growth, provision for additional capacity must be made periodically. The size of the new units to be installed will be governed by joint consideration of: (1) the future rate of load growth; (2) the size of the units available; (3) the urge to standardize on size and type of equipment to simplify repair and reduce inventory on repair parts; (4) varying economies to be obtained with variation in capacity; and (5) the effect on reserve capacity requirements.

Capacity will always have to be installed in advance of the need to maintain load protection. In Fig. 33-2 as an example, the lower curve indicates the trend of the peak load to be expected annually with time. Then, assuming that units to be added have been standardized as to capacity and that a fixed number are to be maintained as installed reserve regardless of the magnitude of the system peak load, the minimum required capacity is found by adding the reserve requirement to the peak load. Since the capacity is installed in units of greater magnitude than the annual increments of capacity required, the installed capacity curve will be stepped in shape. The triangular areas enclosed by the minimum-capacity curve and the installed-capacity curve represent excess capacity that has no immediate use, but this condition cannot be avoided.

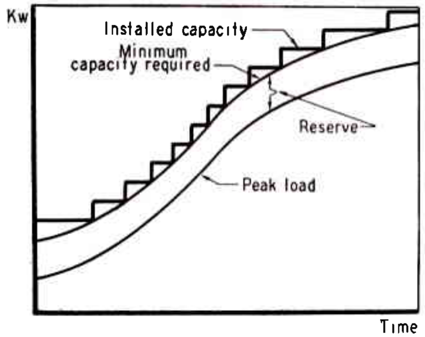

FIG. 33-2. Variation of minimum required capacity and installed capacity for a growing system peak load, assuming that all added units are of equal capacity.

When the growth in load shows that a new unit is required at a certain time, preparations of engineering study and design, the letting of bids, and the actual construction may easily require [illegible] years. Continuous studies should be carried on to provide sufficient capacity for uninterrupted operation. This long period of preparation may at times of serious business recessions cause the utility system to be burdened with capacity that is not being used. Fixed charges must be met for equipment that is temporarily nonproducing, with the possible result of operating at a loss.

33-6. Decreasing Output with Age. In the expansion of a system plant containing a considerable number of units the newer units usually will be the most efficient thermally and preferred for operation. The new units run at a high capacity factor at the expense of less generation on the older units, which will be operated mostly at the time of the peak load. This situation is shown in Fig. 33-3 where two annual load-duration curves have an equal division of load between units of the same size. The second curve is for a period several years later than the first one and shows a considerable growth in peak load and total energy production requiring the addition of five units. In both curves the spinning reserve is maintained equal to one unit. In the first curve, unit E is base-loaded; in the second, it is active for fewer hours and, therefore, generates much less total energy because it is surpassed in efficiency by the newer units F to J. The effect of reduced generation is very marked on units A, B, and C, which operate for much shorter periods as the load grows.

It is sometimes suggested that older and relatively inefficient reserve plants should be dismantled and replaced by new efficient capacity. In such cases it will always be found that the inevitable march of the units to the top of the load curve when they are required for a few hours of operation annually has not been recognized. It is safe to predict that, in a growing system, mere replacement of old reserve capacity by new efficient capacity, solely for the sake of increased efficiency, can seldom, if ever, be economically justified because of the limited operation required for peak-load service.

In Fig. 33-3 the units are assumed to be available 100 per cent of the time. This is seldom true, since equipment must be taken out of service periodically for maintenance and inspection; it also happens that on occasion a unit of equipment is forced out of service by failure of some part of its structure. Evaluation of the availability of equipment for service is expressed by means of the availability factor, which is defined as follows:

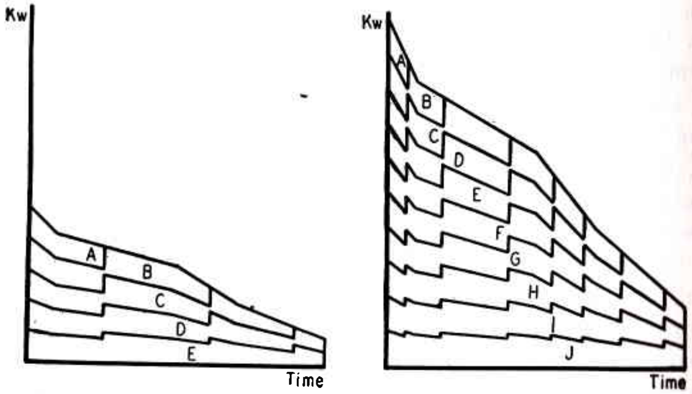

FIG. 33-3. Theoretical comparison of two annual load-duration curves for a rapidly growing system, showing decreasing generation of older units as new efficient units are added to system, maintaining a one-unit reserve. All units of equal capacity.

$$\text{Availability factor} = \frac{\text{hr available for service}}{\text{total hr in period}} \times 100$$

Hours available for service are those hours when the machine is in actual production as well as the hours when the machine is idle but capable of producing. A year is normally the period over which the availability of a machine is measured. Values of the availability factor usually vary between 80 and 90 per cent for steam and hydraulic turbines, steam generators, and oil engines.

By taking account of the availability of the individual units as illustrated in Fig. 33-3 it will be evident that the base-load units in each curve will not occupy the entire base of the curve as shown. During the time when the base-load units are not available, the units next above will serve in their place; thus the generation of the most efficient units will not be quite so great as indicated in the figure, and the generation of the less efficient units will be somewhat larger than indicated.

Variation of the annual capacity factor of an individual unit of equipment serving a system with a growing load is shown in Fig. 33-4. In the early years of its life the unit serves on the base of the load curve. This results in operation at relatively high capacity factors, the exact values depending upon a host of factors among which are: (1) the rate of load growth, (2) the shape of the load curve, (3) the life of the units, and (4) the availability of all the units in the system. With system load growth and the resulting addition of new units to meet the increased demand, the unit is finally forced off the base of the curve and runs fewer hours per day. As time goes on, the daily use of the unit becomes shorter and shorter until it serves only on the peak with correspondingly very low capacity factor.

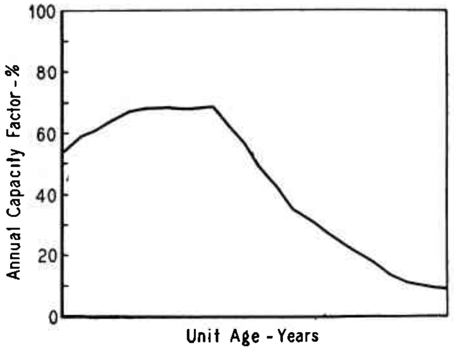

FIG. 33-4. Variation in annual capacity factor of a unit that is serving in a system with a growing load.

33-7. Age Factor. Economic selection of new units for growing systems becomes a special problem. Units are most conveniently compared by evaluating their relative justifiable added investment, i.e.,

$$\Delta P = \frac{C'_o - C''_o}{R}$$

For the given formula the annual operating costs C_o are *uniform* annual costs for the entire life of the machine. With varying use of the machine during its life the operating costs will also vary. Without serious error it may be assumed that the annual operating cost varies directly with the annual capacity factor to be expected of the machine. Then, by computing the operating cost for the first year of each machine, C_{o1}, the costs for succeeding years can be computed by direct proportion.

$$C_{o1}:C_{o2}:C_{o3}: \cdots :C_{on} = \overline{CF}_1:\overline{CF}_2:\overline{CF}_3: \cdots :\overline{CF}_n$$

where $\overline{CF}$ is the annual capacity factor for each year of life that is obtained from a study of expected load growth* and expressed in the form of a graph such as Fig. 33-4. These varying costs must then be reduced to equivalent uniform annual costs. This is done by first finding the present worth W of the series of varying annual costs and then converting it to an equivalent constant annuity A, that is,

$$W = \frac{C_{o1}}{1+i} + \frac{C_{o2}}{(1+i)^2} + \frac{C_{o3}}{(1+i)^3} + \cdots + \frac{C_{on}}{(1+i)^n}$$

where i is the interest rate used in evaluating the machines. Then

$$A = W \frac{i}{1-(1+i)^{-n}}$$

It then follows that

$$\Delta P = \frac{A' - A''}{R}$$

A little study will show that the value of A will be appreciably less than C_{o1}; letting

$$r = \frac{A}{C_{o1}}$$

we find that the age factor r will be less than unity. Then

$$\Delta P = \frac{r}{R}(C'_{o1} - C''_{o1})$$

The advantage of expressing the justifiable added investment in this latter form is to show readily the decrease in value resulting from the reduction in use (corrected for by r) as compared with the value that would obtain if the first year's operating cost held for all years of the life of the machines.

Example 33-4. It has been determined for a system experiencing a steady load growth that the next generating unit to be added will have an annual capacity factor of 70 per cent for the first 5 years, 60 per cent for the next 5 years, 40 per cent for the next 5 years, and 20 per cent for the next 5 years. Machine life is to be taken as

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* For an outline of this method see B. G. A. Skrotzki, Conditions Controlling the Economic Selection of Prime Movers, *Transactions of the AIEE*, 1944.

0 years. Three units having the characteristics shown in the following tabulation are being considered:

Unit	Investment	Annual operating cost, first year
A	$520,000	$96,000
B	550,000	90,000
C	570,000	87,000

Cost of money use is to be taken at 8 per cent; taxes and insurance are 4.5 per cent of investment.

Which unit should be selected, assuming that all are of equal quality and made by firms of equal reliability?

Solution

$$R = \frac{i}{1-(1+i)^{-n}} + (t+j)$$

$$= \frac{0.08}{1-1.08^{-20}} + 0.045 = 0.1469, \text{ or } 14.69\%$$

A method of obtaining the value age factor r shorter than by finding the equivalent constant annual cost A for each unit can be evolved by recognizing that r must be the same for all units, since fundamentally it originates from the values of the annual capacity factors. Thus the values of the annual capacity factors can be taken as the equivalent of the annual operating costs. The present worth of these values is then calculated for a magnitude corresponding to W. Then the equivalent constant annual cost corresponding to A can be found from the value of W. Age factor r will then be the ratio of the value corresponding to A divided by the annual capacity factor for the first year. This method of calculation is used in Table 33-9.

Equivalent uniform annuity:

$$A = 5.418 \times \frac{0.08}{1-1.08^{-20}} = 0.552$$

$$r = \frac{A}{C_{o1}} = \frac{0.552}{0.700} = 0.8$$

Actual additional investment required for unit B above unit A:

$$550{,}000 - 520{,}000 = \$30{,}000$$

Justifiable added investment for unit B above unit A:

$$\Delta P = \frac{r}{R}(C_{o1A} - C_{o1B}) = \frac{0.8}{0.1469} \times (96{,}000 - 90{,}000) = \$32{,}700$$

Hence unit B is the optimum of the two units.

Actual additional investment required for unit C above unit B:

$$570{,}000 - 550{,}000 = \$20{,}000$$

Justifiable added investment for unit C above unit B:

$$\Delta P = \frac{0.8}{0.1469} \times (90{,}000 - 87{,}000) = \$16{,}300$$

Hence unit B is the optimum of the two units, as well as of the three units, and would be the economic choice, all other factors being equal.

TABLE 33-9. DETERMINATION OF EQUIVALENT UNIFORM ANNUAL CAPACITY FACTOR

Year (a)	Annual capacity factor (b)	$\frac{1}{(1+i)^n}$ (c)	Present worth of (b) (d) = (b) × (c)
1	70	0.926	0.649
2	70	0.857	0.600
3	70	0.794	0.555
4	70	0.735	0.515
5	70	0.681	0.477
6	60	0.632	0.379
7	60	0.584	0.350
8	60	0.540	0.324
9	60	0.500	0.300
10	60	0.463	0.278
11	40	0.429	0.172
12	40	0.397	0.159
13	40	0.368	0.147
14	40	0.341	0.136
15	40	0.315	0.126
16	20	0.292	0.058
17	20	0.270	0.054
18	20	0.250	0.050
19	20	0.232	0.046
20	20	0.215	0.043
Total	. .	. . . . .	5.418 = W

If the characteristic of decreasing use with age had not been recognized, unit C would have been selected as the optimum since the justifiable added investment would have been \$20,400.

For most systems the age factor r will remain at the same value for considerable periods until some change in load-growth rate, or change in load-curve shape, or change in cost of money use takes place. Since the value of r depends upon prognostications far into the future, it cannot be regarded as a precise value; at best it is an allowance for a condition that will probably prevail without any accurate knowledge of the exact variations in that condition. For this reason the value of r should generally not be expressed to more than one significant figure.

33-8. Evaluating New Capacity. In the preceding discussion the magnitudes of the operating costs were given to clarify the explanation of the ratio r. To determine the annual operating costs, studies should be made of the amount of energy production the proposed units will assume in relation to the existing equipment. Fuel costs are usually a major portion of the total annual costs of operation; hence pains should be taken in estimating this factor of the total operating costs. An important condition that must be recognized is the effect on fuel consumption of the existing equipment when the new unit is placed in operation. This is illustrated in the following example.

Example 33-5. A system supplied by steam-turbine central stations having an installed capacity of 100,000 kw will have a peak demand of 100,000 kw within 2 years. This system has experienced a steady growth of load in the past, and extensive studies indicate that the rate of growth will be maintained indefinitely into the future. The age factor r to be used in evaluations of additional capacity is 0.7. The capacity of the largest existing units is 10,000 kw, and it has been decided that the new proposed unit shall be of the same capacity to maintain a one-unit installed reserve. The new unit will operate at a higher pressure than the existing turbines and will be fed by a single steam generator.

The estimated load duration that will prevail when the new unit is needed is shown in the following tabulation:

Kw	*Hr*
100,000	100
80,000	1,000
60,000	2,000
40,000	3,000
20,000	2,660

The total system input-output characteristic of the existing units is given by

$$I = 10^6 \times (180 + 10L + 0.001L^3)$$

There are two alternate units being considered for installation, A and B. The input-output characteristic for A is

$$I = 10^6 \times (11 + 8.5L + 0.03L^2 + 0.015L^3)$$

For B it is

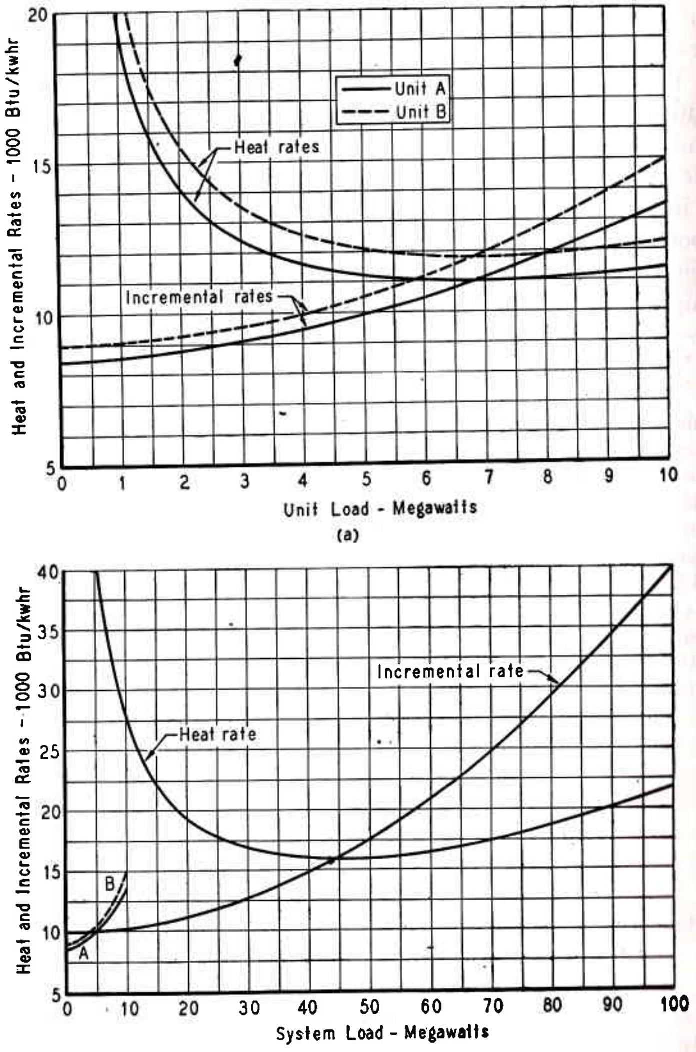
$$I = 10^6 \times (13 + 9L + 0.02L^3)$$

where I is in Btu per hour and L is in megawatts.

The heat and incremental rates are plotted for the three given characteristics in Fig. 33-5a and b.

The unit installed-capacity costs for the proposed units are:

A	\$136 per kw
B	\$128 per kw



(a)

(b)

FIG. 33-5. Heat and incremental rates of existing system and proposed units.

Fuel cost is 30 cents per million Btu, and the fixed charge rate is 15 per cent. Maintenance, labor, and supply costs of the existing and proposed units will be equal to 40 per cent of the fuel costs. Availability factors for new units will be 85 per cent.

Solution. The annual load-duration data are set up in tabular form, and the load is divided between the existing units and each of the proposed units in turn by means of the incremental rates. Fuel requirements are then calculated as shown in Tables 33-10 and 33-11.

TABLE 33-10. LOAD ALLOCATION AMONG UNITS AND FUEL REQUIREMENT

Load, mw	Hours	Million kwhr	With unit *A*						Existing system only	
			Load, mw		Input rate, 10^6 Btu per hr		Total input 10^9 Btu		*I*, 10^6 Btu per hr	Total input 10^9 Btu
			System	*A*	System	*A*	System	*A*		
100	100	10.0	90	10	1809	114.0	181	11.4	2180	218
80	1,000	80.0	70	10	1223	114.0	1223	114.0	1493	1493
60	2,000	120.0	50	10	805	114.0	1610	228.0	996	1992
40	3,000	120.0	31	9	519	100.9	1557	302.7	644	1932
20	2,660	53.2	14	6	325	66.3	866	176.3	388	1032
Total....	8,760	383.2	..	..	...	.....	5437	832.4	....	6667
							6269.4			

Annual system load factor = $(383.2 \times 10^6)/(8{,}760 \times 100{,}000) = 0.437$, or 43.7%.

TABLE 33-11. LOAD ALLOCATION AMONG UNITS AND FUEL REQUIREMENT WITH UNIT *B*

Load, mw	Hours	Load, mw		Input rate, 10^6 Btu per hr		Total input annually, 10^9 Btu		
		System	*B*	System	*B*	System	*B*	Total
100	100	90	10	1809	123.0	181	12.3	
80	1,000	70	10	1223	123.0	1223	123.0	
60	2,000	50	10	805	123.0	1610	246.0	
40	3,000	32	8	533	95.2	1599	285.6	
20	2,660	15	5	336	60.5	895	161.0	
Total	.....	..	..	....	.....	5508	827.9	6335.9

From Table 33-10:

1. Annual output of *A* at 100 per cent availability,

$$(10 \times 3{,}100) + (9 \times 3{,}000) + (6 \times 2{,}660) = 73{,}960 \text{ mwhr}$$

2. Annual output of existing system with *A* at 100 per cent availability,

$$383{,}200 - 73{,}960 = 309{,}240 \text{ mwhr}$$

3. Average heat rate of *A*,

$$\frac{832.4 \times 10^9}{73.96 \times 10^6} = 11{,}270 \text{ Btu per kwhr}$$

4. Actually any individual unit cannot be available 100 per cent of the time. So item 1 must be corrected for the expected availability of 85 per cent. Annual output of A at 85 per cent availability,

$$73{,}960 \times 0.85 = 62{,}870 \text{ mwhr}$$

5. Annual output of existing system with A at 85 per cent availability,

$$383{,}200 - 62{,}870 = 320{,}330 \text{ mwhr}$$

6. It may reasonably be assumed that the average heat rate of the new unit will be the same regardless of the availability of the unit. Then input to A at 85 per cent availability,

$$62.87 \times 10^6 \times 11{,}270 = 708 \times 10^9 \text{ Btu}$$

7. The average heat rate of the existing system will not be the same with A at 100 per cent and 85 per cent availability. For the incremental generation in the existing system from 309,240 to 320,330 mwhr an incremental input must be estimated. This is most conveniently done by utilizing the computation in the last column of the table and calculating an average incremental heat rate as follows:

$$\frac{(6{,}667 - 5{,}437) \times 10^9}{(383.2 - 309.2) \times 10^6} = 16{,}620 \text{ Btu per kwhr}$$

8. Annual input to system with A at 85 per cent availability,

$$(5{,}437 \times 10^9) + (320.33 - 309.24) \times 10^6 \times 16{,}620 = 5{,}621.3 \times 10^9 \text{ Btu}$$

9. Annual operating cost of A,

Fuel = $708 \times 10^9 \times 0.30/10^6$	$212,400	
Labor, maintenance, and supplies, 40 per cent of fuel cost	85,000	
Total operating cost		$ 297,400

10. Annual operating cost of existing system,

Fuel = $5{,}621 \times 10^9 \times 0.30/10^6$	$1,686,300	
Labor, maintenance, and supplies, 40 per cent of fuel cost	674,500	
Total operating cost		$2,360,800

Following the same scheme of computation for B from Table 33-11 as for A from Table 33-10,

1. Annual output of B at 100 per cent availability,

$$(10 \times 3{,}100) + (8 \times 3{,}000) + (5 \times 2{,}660) = 68{,}300 \text{ mwhr}$$

2. Annual output of existing system with B at 100 per cent availability,

$$383{,}200 - 68{,}300 = 314{,}900 \text{ mwhr}$$

3. Average heat rate of B,

$$\frac{827.9 \times 10^9}{68{,}300{,}000} = 12{,}120 \text{ Btu per kwhr}$$

4. Annual output of B at 85 per cent availability,

$$68{,}300 \times 0.85 = 58{,}100 \text{ mwhr}$$

5. Annual output of existing system with B at 85 per cent availability,

$$383{,}200 - 58{,}100 = 325{,}100 \text{ mwhr}$$

6. Input to B at 85 per cent availability,

$$58.1 \times 10^6 \times 12{,}120 = 705 \times 10^9 \text{ Btu}$$

7. Average increment heat rate for system (from Table 33-10),

$$16{,}620 \text{ Btu per kwhr}$$

8. Annual input to system with B at 85 per cent availability,

$$(5{,}508 \times 10^9) + (325.1 - 314.9) \times 10^6 \times 16{,}620 = 5{,}677.4 \times 10^9 \text{ Btu}$$

9. Annual operating cost of B,

Fuel = $705 \times 10^9 \times 0.30/10^6$	$211,500	
Labor, maintenance, and supplies, 40 per cent of fuel cost	84,600	
Total operating cost		$ 296,100

10. Annual operating cost of existing system,

Fuel = $5{,}677 \times 10^9 \times 0.30/10^6$	$1,703,100	
Labor, maintenance, and supplies, 40 per cent of fuel cost	681,200	
Total operating cost		$2,384,300

By comparing the operating costs of A and B, it will be seen that they differ by only $1,300 (items 9). However, the effect on the existing system's operating costs is much more, being $23,500. It is evident that *all* changes in operating costs must be included properly to evaluate the units relative to each other. Then the justifiable added investment for A over B is

$$P = \frac{r}{R}(C_{o1B} - C_{o1A})$$

$$= \frac{0.70}{0.15}(296{,}100 + 2{,}384{,}300) - (297{,}400 + 2{,}360{,}800) = \$103{,}600$$

The actual required investment for A over B is

$$(136 - 128) \times 10{,}000 = \$80{,}000$$

Hence A is the economic choice; its greater efficiency justifies the added investment.

As a check the comparison can be made on the basis of total annual costs as in Table 33-12. Only the annual operating costs need be considered relative to each

TABLE 33-12. RELATIVE ECONOMY OF UNITS *A* AND *B*

Unit	Unit cost	Total investment	C_f	Relative C_o	Relative C
A	$136	$1,360,000	$204,000	0	$204,000
B	128	1,280,000	192,000	$15,500	207,500

other in this comparison. If the operating costs of the plan with *A* are taken as the base, i.e., as relatively zero, then the relative costs for the plan with *B* will be $r(C_{o1B} - C_{o1A})$. The age factor r must be included to account properly for the reduction in use, with age, of the units. The relative total annual costs for *A* are less than for *B*; hence the former is the economic unit. The same conclusion was obtained with the justifiable-added-investment method.

33-9. Hydraulic-station Economics. The ability of a hydraulic station to meet the system peak-load demand, as determined by the available stream flow and storage or pondage, is a measure of its *firm* capacity to produce primary power. To be useful to a given system, the hydraulic plant must have the same capacity availability as a steam station at the time of the system peak load. Therefore the minimum stream flow to be expected at such a time is the prime factor in determining the firm capacity of the plant. Thus, in Fig. 32-18*b*, if the load-duration curve refers to the peak-load week of the plant and the shaded area represents the hydraulic output available in stream flow and pondage, the intercept *b-c* then measures the firm capacity of the plant.

The advisability of installing a hydraulic plant is always measured by a comparison of its costs against the costs for an alternate steam or internal-combustion (i-c) plant. As an example of such an investigation, consider the problem of Sec. 33-8, where the system had a load curve with an annual load factor of 43.8 per cent for which the steam-turbine unit *A* was the economic choice. Another alternative exists in a hydraulic site where studies of stream flow (Figs. 25-1 and 25-2) and available pondage indicate that the plant can produce a firm capacity of 10,000 kw. For a year of average flow conditions other studies show that the plant with a 10,000-kw turbine installed can at 85 per cent availability generate 52,600,000 kwhr. This generation will save the fuel required to generate an equal output in the existing steam stations.

In Table 33-10 it was shown that the total annual input to the existing system was 6,667 × 10⁹ Btu to generate the requirements of the anticipated load curve of 383,200,000 kwhr. To estimate the fuel savings by hydraulic generation, it is necessary to determine the differential unit fuel cost. This is most closely estimated by taking the existing system input (when operating with unit *B*) from Table 33-11, which is $5{,}508 \times 10^9$ Btu. The corresponding generation of the existing units is determined by integrating the generation and corresponding hours in the existing units, which were found to add to 314,900,000 kwhr. Then the differential unit fuel saving or cost at 30 cents per million Btu will be

$$\frac{(6{,}667 - 5{,}508) \times 10^9 \times 30}{(383.2 - 314.9) \times 10^6 \times 10^6} = 0.509 \text{ cent per kwhr}$$

Then the total fuel savings effected by the hydraulic unit will be

$$52{,}600{,}000 \times 0.00509 = \$268{,}000, \text{ annually}$$

To serve the expected-load curve, the existing system incurs an annual fuel cost of $6{,}667 \times 10^9 \times 0.30/10^6 = \$2{,}000{,}000$ (see Table 33-10). With unit *A* generating 62,870,000 kwhr the total fuel cost is

$$212{,}400 + 1{,}686{,}300 = \$1{,}898{,}700$$

Thus unit *A* effects an annual fuel saving of

$$2{,}000{,}000 - 1{,}898{,}700 = \$101{,}300$$

Though unit *A* generates a greater output than the hydraulic station, its fuel saving is less because it requires fuel for its own operation.

The hydraulic station can be installed complete with all accessories and necessary transmission lines for $324 per kilowatt of capacity. General experience seems to indicate that the fixed-charge rate on hydraulic equipment can be assumed lower than for a steam plant, because: insurance is less owing to the nature of the equipment; taxes are lower owing to the rural location; the aggregate life is probably longer than that of steam equipment and thus lowers the required depreciation rate. The fixed-charge rate for the hydraulic station is taken at 13 per cent compared with 15 per cent for the steam plant. Operating labor, maintenance, and supplies are usually more for a steam plant than for an equal-sized hydraulic plant. The high temperature experienced by steam equipment causes higher maintenance. The separate phases of steam-plant operation need more men than the hydraulic plant.

While thermal-unit service travels toward the peak of the load-duration curve as the unit grows older, hydraulic stations always serve the base of the curve as long as water is available, regardless of their age.

Comparing new steam unit *A* and the hydraulic plant, we get:

1. Total system energy requirement	383,200 mwhr	
2. Annual system load factor, per cent	43.8	
Steam unit *A* of 10-mw capacity:		
3. Annual generation	62,870 mwhr	
4. Total investment at $128 per kw	$1,280,000	
5. Fixed charges at 15 per cent		$192,000
6. New-unit operating labor, maintenance, and supplies		85,000
7. Total new costs		$277,000
8. Fuel saving at 30 cents per million Btu = $101,200 × 0.7[a]	$70,900	
9. Saving in labor, maintenance and supplies on existing system at 40 per cent of fuel saving = 0.4 × (2,000,000 − 1,686,300) × 0.7[a]	87,800	
10. Total net annual savings		158,700
11. Net annual total cost		$118,300
Hydraulic station of 10-mw capacity:		
12. Annual generation	52,600 mwhr	
13. Total investment at $324 per kw	$3,240,000	
14. Fixed charges at 13 per cent		$421,200
15. New-station labor, maintenance, and supplies		40,400
16. Total new costs		$461,600
17. Fuel saving at 30 cents per million Btu	$268,000	
18. Saving in labor, maintenance, and supplies on existing system at 40 per cent of fuel saving = 0.4 × $268,000	107,200	
19. Total net annual savings		$375,200
20. Net annual total cost		$ 86,400

[a] $r = 0.7$ to correct first year's savings to equivalent annual saving for life of unit.

The hydraulic plant is the better selection, entailing $31,900 less in annual expenditure. The fuel savings outweigh the high fixed charges of the hydraulic plant as compared with the steam unit. Calculation will show that the hydraulic investment can rise to $348 per kilowatt of capacity to show parity with the steam plant. For any higher unit cost the steam plant would be the economic selection, all other factors being equal.

Often the major portion of the cost of developing a water-power site lies in the dam, the acquisition of land for the storage reservoir, flumes, tunnels, highway and railroad relocation, and other items that are more or less independent of the station capacity. When sufficient water flow is available, the installation of additional water-turbine units over the firm capacity often may be accomplished at unit capacity costs of about $80 per kilowatt since the primary structural costs have already been met. This capacity can generate secondary power at times of high water flow. At such times all or part of the thermal-station capacity of the system may be shut down to save fuel. When it is installed solely for this type of operation, the fixed charges on the secondary-power equipment must be less than the fuel savings at the thermal stations. Secondary capacity is often installed when an industry like the wood-pulp paper mills that can operate on a "when, as, and if" basis is supplied by the system. The mills operate when secondary power is available and shut down otherwise. Secondary power can be sold at much lower rates than primary power.

By far the major portion of the annual costs of hydraulic-station operation is the fixed charges. Hence, if the station operates at a low capacity factor, the unit energy cost will be high, but as capacity factor increases, the unit energy costs steadily drop to a minimum at full loading.

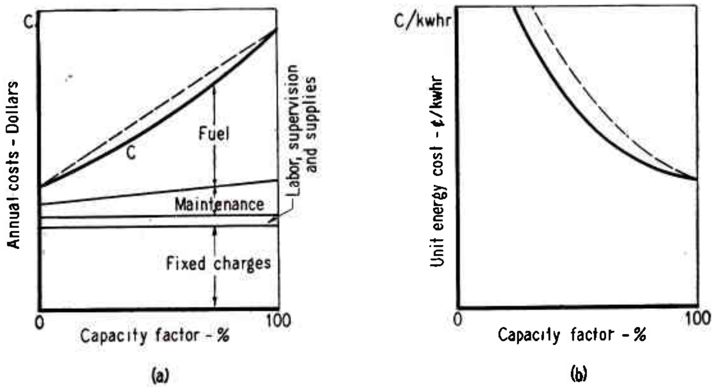


Fig. 33-6. Variation with station capacity factor of (*a*) total annual costs and (*b*) unit energy cost.

There is no reason why hydraulic stations should be considered inherently superior to, or producers of power at lower costs than, thermal stations. Local circumstances alone demonstrate which type is more suitable. Many systems attain maximum economic operation by having both types included as sources of energy.

33-10. Variation of Unit Energy Cost with Capacity Factor. The unit cost of energy produced by a station depends upon the degree of use of the plant, i.e., upon its annual capacity factor (Fig. 33-6*a*). The variation of annual cost with capacity factor is made up of the components indicated. The fixed costs remain constant for all values of capacity factor. Thus, at low capacity factors, the fixed charges are shared among a smaller number of units of energy and result in relatively high unit energy costs. In contrast, at high factors the same total fixed charges are shared by a large number of units of energy to reduce the unit energy cost. Labor, supervision, and supplies will often remain substantially constant for all

TABLE 33-13. CALCULATION OF UNIT ENERGY COSTS OF 10,000-KW STEAM STATION

	Capacity factor								
	0	10	20	30	40	50	60	80	100
Load, mw	0	1	2	3	4	5	6	8	10
Input, 10^6/Btu per hr	11	19.54	28.24	37.18	46.44	56.13	66.32	88.60	114.00
HR, Btu per kwhr	∞	19,550	14,120	12,390	11,610	11,230	11,050	11,080	11,400
Annual mwhr	0	8,760	17,520	26,280	35,040	43,800	52,560	70,080	87,600
Annual costs:									
Fixed charges	$150,000	$150,000	$150,000	$150,000	$150,000	$150,000	$150,000	$150,000	$150,000
Labor, supervision, and supplies	30,000	30,000	30,000	30,000	30,000	30,000	30,000	30,000	30,000
Maintenance	1,000	4,000	7,000	10,000	13,000	16,000	19,000	25,000	31,000
Fuel	19,300	34,300	49,500	65,200	81,300	98,500	116,200	155,300	199,700
Total cost:									
Min	$200,300	$218,300	$236,500	$255,200	$274,300	$294,500	$315,200	$360,300	$410,700
Max	200,300	221,340	242,380	263,420	284,460	305,500	326,540	368,620	410,700
Unit energy cost, cents per kwhr:									
Fixed charges	∞	1.71	0.86	0.57	0.43	0.34	0.28	0.21	0.17
Labor, supervision, and supplies	∞	0.34	0.17	0.11	0.08	0.07	0.06	0.04	0.04
Maintenance	∞	0.05	0.04	0.04	0.04	0.04	0.04	0.04	0.03
Fuel	∞	0.39	0.28	0.25	0.23	0.22	0.22	0.22	0.23
Total cost:									
Min	∞	2.49	1.35	0.97	0.78	0.67	0.60	0.51	0.47
Max	∞	2.52	1.38	1.00	0.81	0.70	0.62	0.53	0.47

In actual practice it is not generally possible to achieve the ultimate minimum cost at 100 per cent capacity factor since the equipment must be taken out of service periodically for inspection, cleaning, and overhaul.

rates of operation, though some stations show a slight increase with increasing generation. Maintenance and repairs increase with generation, most of the increase being caused by steam-generator deterioration with service. The fuel cost varies directly with the capacity factor. The reason for the envelope in Fig. 33-6*a* will be recalled by referring to Fig. 32-5.

The two limits of the envelope define the region within which the total annual costs may lie for the various average capacity factors. The unit energy cost at any given capacity factor is then determined in Fig. 33-6*b* as follows:

$$\text{Unit energy cost} = \frac{\text{total annual cost}}{\text{station capacity} \times \dfrac{\text{capacity factor}}{100} \times 8{,}760}$$

The unit-energy-cost curve also appears in the form of an envelope, the precise average for any year depending upon the shape of the load-duration curve.

In general, the unit energy cost will be a minimum when the generating facilities are used to the maximum extent. The following example shows the chief reason for the aggressive sales effort made by utility companies to load their generating stations to the maximum.

Example 33-6. Determine the unit energy costs for a single-unit steam-turbine station of 10 mw capacity with the input-output characteristic of

$$I = 10^6 \times (11 + 8.5L + 0.03L^2 + 0.015L^3)$$

where I is in Btu per hour and L is in megawatts. Installed capacity cost is $100 per kilowatt, and the fixed-charge rate is 15 per cent. Fuel costs 20 cents per million Btu. Labor, supervision, and supplies cost $30,000 annually. Maintenance varies linearly from $1,000 annually with the station hot but not generating to $31,000 annually with the station operating continuously at maximum load.

Find the minimum and maximum unit energy costs, assuming that the station is always hot, for capacity factors of 0, 10, 20, 30, 40, 50, 60, 80, and 100 per cent.

Solution. Table 33-13 contains the entire solution.

33-11. Incremental Fuel Cost. In economic evaluations there is a great tendency to resort to average rates to arrive at a total cost. Extreme caution should be exercised in doing this, especially in dealing with unit fuel cost of energy produced by a generating station. Thus, if the average unit fuel cost over a year is 0.3 cent per kilowatthour for a given output on a station, there is a strong temptation to assume that any additional generation at the same station can be produced at the same unit fuel cost. The additional fuel cost for the additional generation depends entirely upon the manner in which the generation is added. This is demonstrated in the following example.

Example 33-7. A low-pressure steam station of 10,000-kw capacity has the following input-output characteristic,

$$I = 20 + 8L + 0.2L^2 + 0.06L^3$$

where I is in millions of Btu per hour and L is in megawatts. This station carries all existing daily load as listed in Table 33-14 and illustrated in Fig. 33-7. Find the average heat rate of an additional output of 24,000 kwhr daily when added as follows:

1. Equally over the 24 hr, i.e., raising the load by 1 mw each hour, as illustrated in Fig. 33-7*b*.
2. Filling in the morning valley, as illustrated in Fig. 33-7*c*.
3. Adding generation to the maximum possible during the peak period, as illustrated in Fig. 33-7*d*.

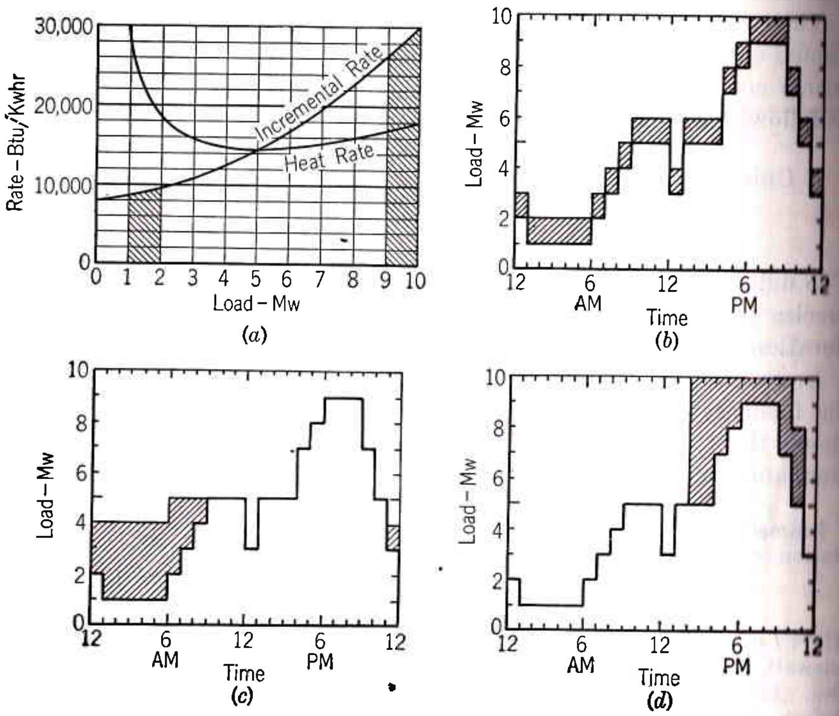


FIG. 33-7. Figures for demonstrating variation in incremental fuel cost for a *constant total* added generation.

The shaded areas on the load curves are all equal to 24,000 kwhr. Compare the incremental heat rates of the incremental generation for each case with the total average heat rate for the entire output.

Solution. Table 33-15 lists the input rates, heat rates, and incremental rates vs. the loads for the station. The heat- and incremental-rate curves are given in Fig. 33-7*a*.

TABLE 33-14. EXISTING STATION LOAD

Hour ending	Load, mw	Hour ending	Load, mw
1 A.M.	2	1 P.M.	3
2	1	2	5
3	1	3	5
4	1	4	5
5	1	5	7
6	1	6	8
7	2	7	9
8	3	8	9
9	4	9	9
10	5	10	7
11	5	11	5
12 M.	5	12 P.M.	3

TABLE 33-15. STATION CHARACTERISTICS

Load, mw	Input, 10^6 Btu per hr	Heat rate, Btu per kwhr	Incremental rate, Btu per kwhr
0	20.00	∞	8000
1	28.26	28,260	8580
2	37.28	18,640	9520
3	47.42	15,807	10,820
4	59.04	14,760	12,480
5	72.50	14,500	14,500
6	88.16	14,693	16,880
7	106.38	15,197	19,620
8	127.52	15,940	22,720
9	151.94	16,882	26,180
10	180.00	18,000	30,000

In Table 33-16 the daily inputs are computed for each of the daily load schedules or cases.

For the daily generation of 106,000 kwhr of the existing load the integrated or daily average heat rate is (1,720.76 × 10^6)/106,000 = 16,230 Btu per kwhr. The daily average load is 106,000/24 = 4,410 kw. As a matter of interest, the instantaneous heat rate for a constant load of this value on the station is 14,600 Btu per kwhr.

With 24,000 kwhr generation added at the rate of 1,000 kw per hr the daily average heat rate is 2,094,960,000/130,000 = 16,115 Btu per kwhr. This compares with the average heat rate of the existing load alone of 16,230 Btu per kwhr, an improve-

Table 33-16. Calculation of Daily Station Inputs

Hour ending	Existing load		24 mwhr added to existing output					
			1 mw per hr		At minimum loads		At maximum loads	
	Mw load	Input, 10^6 Btu	Mw load	Input, 10^6 Btu	Mw load	Input, 10^6 Btu	Mw load	Input, 10^6 Btu
1 A.M.	2	37.28	3	47.42	4	59.04	2	37.28
2	1	28.26	2	37.28	4	59.04	1	28.26
3	1	28.26	2	37.28	4	59.04	1	28.26
4	1	28.26	2	37.28	4	59.04	1	28.26
5	1	28.26	2	37.28	4	59.04	1	28.26
6	1	28.26	2	37.28	4	59.04	1	28.26
7	2	37.28	3	47.32	5	72.50	2	37.28
8	3	47.42	4	59.04	5	72.50	3	47.42
9	4	59.04	5	72.50	5	72.50	4	59.04
10	5	72.50	6	88.16	5	72.50	5	72.50
11	5	72.50	6	88.16	5	72.50	5	72.50
12 M.	5	72.50	6	88.16	5	72.50	5	72.50
1 P.M.	3	47.42	4	59.04	3	47.42	3	47.42
2	5	72.50	6	88.16	5	72.50	5	72.50
3	5	72.50	6	88.16	5	72.50	10	180.00
4	5	72.50	6	88.16	5	72.50	10	180.00
5	7	106.38	8	127.52	7	106.38	10	180.00
6	8	127.52	9	151.94	8	127.52	10	180.00
7	9	151.94	10	180.00	9	151.94	10	180.00
8	9	151.94	10	180.00	9	151.94	10	180.00
9	9	151.94	10	180.00	9	151.94	10	180.00
10	7	106.38	8	127.52	7	106.38	10	180.00
11	5	72.50	6	88.16	5	72.50	8	127.52
12 P.M.	3	47.42	4	59.04	4	59.04	3	47.42
Total.....	106	1720.76	130	2094.96	130	1981.80	130	2274.68

ment in the efficiency of generation. The added energy input can be evaluated in terms of the average incremental heat rate, which is

$$\frac{2{,}094{,}960{,}000 - 1{,}720{,}760{,}000}{130{,}000 - 106{,}000} = 15{,}590 \text{ Btu per kwhr}$$

The average incremental heat rate for this condition is 3.9 per cent less than the daily average heat rate of the existing load.

With 24,000 kwhr generation added to fill the load-curve valley the daily average heat rate becomes 1,981,800,000/130,000 = 15,245 Btu per kwhr, as compared with 16,230 Btu per kwhr for the existing load. For this method of adding generation the average incremental heat rate is

$$\frac{1{,}981{,}800{,}000 - 1{,}720{,}760{,}000}{130{,}000 - 106{,}000} = 10{,}875 \text{ Btu per kwhr}$$

This is 33 per cent less than the existing-load daily average heat rate.

With 24,000 kwhr generation added at the peak period of the load curve the daily average heat rate becomes 2,274,680,000/130,000 = 17,500 Btu per kwhr as compared with 16,230 Btu per kwhr for the existing load. For this manner of addition the average incremental heat rate is

$$\frac{2{,}274{,}680{,}000 - 1{,}720{,}760{,}000}{130{,}000 - 106{,}000} = 23{,}080 \text{ Btu per kwhr}$$

This is 42.2 per cent greater than the existing daily average heat rate. Summarizing:

Load conditions	Average heat rate, Btu per kwhr	Average incremental heat rate, Btu per kwhr	Variation of average *IR* from existing load *HR*, per cent
Existing load......................	16,230		
24 mwhr added, 1 mw per hr	16,115	15,590	− 3.9
24 mwhr added at minimum loads ...	15,245	10,875	−33.0
24 mwhr added at maximum loads	17,500	23,080	+42.2

The fallacy of using average rates without precise knowledge of how the load is added becomes readily apparent in this example. The variation in average incremental heat rate for the same block of energy on the same base-load curve is more than 112 per cent.

In this example the average incremental heat rates were computed from the total inputs. They can also be computed from the incremental heat rate characteristic of the station. The areas under the incremental-rate curve (see Fig. 33-7*a*) are the incremental inputs for the given blocks of added load at each hour. As illustrated, the input added for adding 1 mw of load above a base load of 1 mw is the hatched area between 1 and 2 mw. Similarly, when 1 mw of load is added to a base of 9 mw, the added input is the area hatched between 9 and 10 mw. By properly calculating these areas, adding them, and dividing by the total added output, the average incremental rate will be determined, which should check exactly the values obtained from using the total inputs.

33-12. Replacement of Existing Equipment. Obsolescence of existing equipment is rarely apparent. When it is suspected to exist, only a careful analysis of all costs involved—those of the existing equipment and those of the new, more efficient equipment—can determine the answer. A condition that causes much confusion in obsolescence analysis is the proper interpretation to be placed on the present value of the existing equipment. On the books of the owner there will be a certain "book value" placed on the equipment. This is found by first entering the cost of the equipment when new and subtracting therefrom an arbitrary amount each year to indicate the estimated amount of recovered capital investment; the remainder is the book value and is merely the estimate of the unrecovered investment at any particular date.

Fundamentally the capital investment in the existing equipment is a past expenditure. In economic-evaluation parlance it is commonly termed a "sunk" cost. A sunk cost has no bearing on future monetary expenditures. The sole question to be answered in an obsolescence analysis is: Will the future annual expenditures for the life of the proposed new equipment be less than the future annual expenditures incurred by the existing equipment if it is kept in operation?

If the existing equipment has been bought on the proceeds of bond sales, it is true that annual payments would have to be met for interest and amortization. However, if the equipment is retired in favor of more efficient equipment before the bonds have matured, these payments will be unaffected and so need not be considered in an obsolescence analysis. This analysis reduces to comparing the *total of fixed and operating charges* of the proposed new equipment with the *operating charges* of the existing equipment.

Example 33-8. A generating station consisting of three 1,000-kw oil engines serves an industrial plant which has a yearly energy demand that is expected to remain the same in the future, as in the following tabulation:

Hr	*Kw*
4,000	2,000
3,000	1,000
1,000	500
760	300

The loads at 1,000 kw and above must be protected by running a one-engine reserve; loads below 1,000 kw need not be protected since at such times short outages due to failures can be tolerated. These existing engines are 3 years old and cost $90 per kilowatt installed. The installation cost is being amortized on the books of the company on the basis of an 8-year life.

The input-output characteristic of these engines is (per engine)

$$I = 2{,}000{,}000 + 8{,}000L + 8L^2 - 0.003L^3$$

where I is in Btu per hour and L is in kilowatts. The lubricating-oil consumption of these engines has been 0.001 gal per kwhr.

A new oil engine of 1,000-kw capacity has since been developed with the following input-output characteristic:

$$I = 1{,}300{,}000 + 7{,}000L + 4.1L^2 - 0.0014L^3$$

The installed cost of this engine will be $85 per kilowatt and the lubricating-oil consumption 0.0006 gal per kwhr. Operating labor costs are the same for the new and the existing engine, but maintenance of the old engines averages $4,000 per year per engine and the new engine is expected to require only $3,000 per year. The existing engines can be sold for a net price of $40,000 each if it should be decided to replace any of them.

Find the number of engines that should be retired and replaced with new ones, if fuel costs 25 cents per million Btu, lubricating oil costs 70 cents per gallon, money use costs 8 per cent, and the life of the new engine is taken as 8 years. Taxes and insurance amount to 4 per cent of the original investment. Salvage value of the new engines will be assumed equal to the cost of removal at the termination of their life. Engine availability is to be taken at 85 per cent.

Solution

TABLE 33-17. ANNUAL STATION ENERGY REQUIREMENTS WITH PRESENT ENGINES

Load, kw	Hours	Mwhr	No. of engines	Input per engine, 10^6 Btu per hr	Total input, 10^6 Btu
2,000	4,000	8,000	3	10.03	120,400
1,000	3,000	3,000	2	7.63	45,800
500	1,000	500	1	7.63	7,630
300	760	228	1	5.04	3,830
Total	8,760	11,728	. .	. . . . .	177,660

TABLE 33-18. ANNUAL STATION ENERGY REQUIREMENTS WITH ONE NEW ENGINE AT 100 PER CENT AVAILABILITY

Load, kw	Load division, kw			Hr	Output, mwhr		Input				
	New engine	Old engines					10^6 Btu per hr per engine		Total inputs, 10^6 Btu		
		No. 1	No. 2		New engine	Old engines	New engine	Old engines	New engine	Old engines	Total
2,000	1,000	500	500	4,000	4,000	4,000	11.00	7.63	44,000	61,100	105,100
1,000	800	200	0	3,000	2,400	600	8.81	3.90	26,400	11,700	38,100
500	500	0	0	1,000	500	0	5.65	0	5,650	0	5,650
300	300	0	0	760	228	0	3.73	0	2,840	0	2,840
Total	. . . . .	. . .	. . .	. . . . .	7,128	4,600	. . . . .	. . . .	78,890	72,800	151,690

Correction of inputs and outputs to 85 per cent availability for new engine:

$$\text{New engine average heat rate} = \frac{78{,}890{,}000{,}000}{7{,}128{,}000} = 11{,}070 \text{ Btu per kwhr}$$

$$\begin{aligned}\text{Average increment heat rates of old engines} &= \frac{(177{,}660 - 72{,}800) \times 10^6}{(11{,}728 - 4{,}600) \times 1{,}000}\\ &= 14{,}720 \text{ Btu per kwhr}\end{aligned}$$

$$\begin{aligned}\text{New engine annual output at 85 per cent availability} &= 7{,}128 \times 0.85\\ &= 6{,}060 \text{ mwhr}\end{aligned}$$

Corresponding old engine output = 4,600 + 7,128 − 6,060 = 5,668 mwhr

New engine input at 85 per cent availability

$$= 6{,}060{,}000 \times 11{,}070 \qquad = 67{,}100 \times 10^6 \text{ Btu}$$

Corresponding old engine input

$$= 72{,}800 + 14{,}720 \times (7{,}128 - 6{,}060) \qquad = 88{,}500 \times 10^6 \text{ Btu}$$

Total station input with new engine at 85 per cent availability $= 155{,}600 \times 10^6$ Btu

$$\text{Sinking-fund depreciation rate} = \frac{0.08}{1.08^8 - 1} = 0.094, \text{ or } 9.4\%$$

TABLE 33-19. COMPARISON OF ANNUAL COSTS OF EXISTING STATION WITH SCHEME REPLACING ONE ENGINE

Annual cost item	Existing station		With one engine replaced	
Fuel oil @ 25¢/million Btu........	177,660 × 10^6 Btu	$44,400	155,600 × 10^6 Btu	$38,000
Lubricating oil @ 70¢ per gal......	11,728,000 × 0.001	8,200	5,668,000 × 0.001	3,970
			6,060,000 × 0.0006	2,540
Maintenance....................	4,000 × 3	12,000	4,000 × 2	8,000
			3,000 × 1	3,000
Taxes and insurance @ 4%.......	1,000 × 3 × 90	10,800	1,000 × 2 × 90	7,200
			1,000 × 1 × 85	3,400
Amortization of new net investment, @ 9.4%................	....	0	(85,000 − 40,000)	4,230
Interest on new net investment @ 8%......................	....	0	(85,000 − 40,000)	3,600
Comparative annual costs.........	....	$75,400	....	$73,940

Annual savings in total costs = 75,400 − 73,940 = $1,460 (Table 33-19)

By replacing one existing engine with a new one the future annual expenditures will be reduced by $1,460. In effect what has happened is that some of the original operating costs have been reduced and part of this reduction has been replaced by the investment charges on the new money of $45,000.

In making the same study for replacing two engines it will be found that no savings result; hence only one unit can justifiably be replaced.

Results of the obsolescence study may also be interpreted on the basis of earnings on investment or rate of return i, solved for by trial and error from

$$\frac{A}{P} = \frac{i}{1 - (1 + i)^{-n}}$$

In this case

$$A = (75{,}400 - 73{,}940) + 3{,}600 + 4{,}230 = \$9{,}290$$

and

$$P = 85{,}000 - 40{,}000 = \$45{,}000 \qquad n = 8 \text{ years}$$

Then

$$\frac{9{,}290}{45{,}000} = \frac{i}{1 - (1 + i)^{-8}} = 0.2062$$

from which $i = 12.7\%$ compared with the desired 8%

The foregoing example is a case of partial obsolescence of equipment caused by advancement of the art. Another type of obsolescence relates to deterioration from use. In this type of problem, to restore the equipment to original or better condition, an investment in rebuilding is required. Such rebuilding ordinarily will not affect tax or insurance costs, but, of course, sufficient savings must be realizable to repay the investment with a suitable return. Here again, sunk costs have no bearing on the problem; the only object is to reduce future annual expenditures. Expected future life of the equipment, as well as its utilization, should be thoroughly investigated to ensure placing the analysis and conclusions on a sound basis. This will be evident from a study of the following example.

Example 33-9. A 20,000-kw steam turbine has been in use for 4 years and generates 105,000,000 kwhr annually. The first year this energy was produced at an average heat rate of 15,000 Btu per kwhr. Owing to blade erosion the average heat rate increases at the rate of 0.05 per cent per 1,000 hr of operation, and this is expected to continue for the life of the machine despite adequate routine maintenance. To reblade the turbine and restore it to its original efficiency will cost $35,000. It has been carefully determined that the total life of the turbine should be taken as 15 years, and it will be required to produce the same amount of energy annually in the future as it has in the past. If fuel costs 20 cents per million Btu, a minimum of 6 per cent return is required of an investment, and the turbine operates 6,000 hr per year, determine when and if the turbine should be rebladed. The rebladed turbine will deteriorate in efficiency at the same rate as at present.

Solution. First year's annual fuel cost:

$$105{,}000{,}000 \times 15{,}000 \times \frac{0.2}{10^6} = \$315{,}000$$

Increase in fuel cost per year:

$$315{,}000 \times 6{,}000 \times \frac{0.0005}{1{,}000} = \$945$$

In the second column of Table 33-20 is the increase in fuel cost from the initial year of operation of the turbine for any year of its life. Hence, if the turbine is rebladed in any one of these years, the subsequent years will realize the fuel savings effected in that year, since the blading will again deteriorate at the same rate. In the third column is the corresponding life that the reblading investment will enjoy if the turbine is rebladed in any given turbine year. The fourth column is the factor for computing the annual interest and amortization of the reblading investment made in any given year. The fifth column gives the product of the factor and the $35,000 investment. To justify the reblading, the savings in fuel from column 2 must be at least equal to or greater than the interest and amortization given in column 5. This occurs in the seventh, eighth, ninth, and tenth years. The greatest difference occurs in the ninth year, where the largest return on the $35,000 reblading investment is indicated.

TABLE 33-20. TURBINE REBLADING EVALUATION

Turbine age, years (n)	Subsequent annual saving when rebladed in given year (n)	Life of reblading investment, years (m)	$\frac{0.06}{1 - 1.06^{-m}}$	Annual interest and amortization of \$35,000 invested in year (n), $i = 0.06$	Rate of return for subsequent annual saving referred to \$35,000 investment, per cent
1	\$ 0	14	0.1076	\$ 3,760	Negative
2	945	13	0.1130	3,960	Negative
3	1,890	12	0.1193	4,180	Negative
4	2,835	11	0.1268	4,440	Negative
5	3,780	10	0.1359	4,750	1.5
6	4,725	9	0.1470	5,150	4.1
7	5,670	8	0.1610	5,640	6.2
8	6,615	7	0.1791	6,270	7.5
9	7,560	6	0.2034	7,120	8.0
10	8,505	5	0.2374	8,300	6.9
11	9,450	4	0.2886	10,100	3.2
12	10,395	3	0.3741	13,100	Negative
13	11,340	2	0.5454	19,100	Negative
14	12,285	1	1.06	37,100	Negative
15	13,230	0			

Desirability of reblading can also be determined by the A/P ratio, where the figures in the second column are the values of A for each year and P is \$35,000. Rates of return so calculated are given in the sixth column. The maximum rate of return of 8.0 per cent occurs with reblading done in the ninth year. This compares with the minimum desired rate of 6 per cent.

Another type of obsolescence problem that occurs quite frequently in growing utility systems is the conversion of firing equipment in old steam stations that are serving as peak-load and stand-by plants. These plants often have stoker-fired boilers and operate only 4 to 6 hr a day during the peak season. When a boiler operates in this manner on successive days for a period of several weeks or months, the greatest economy is realized by banking the fires during the inactive hours instead of permitting them to burn out each day and starting a fresh fire the next day. Banking a fire maintains a slow rate of combustion in the furnace to keep the boiler steam pressure at some value below the working pressure. A fire in this condition can bring a boiler to a steaming state within several minutes, whereas if a fresh fire had to be started, it might take several hours. Maintaining a banked fire consumes fuel that does not produce any useful energy output.

It sometimes proves economical to remove the stoker and replace it with oil-firing equipment, even if oil should have a higher unit price than coal. The savings are brought about by elimination of the nonproducing element of banked-fire fuel; the elimination of the repairs of a stoker which has many parts exposed to more severe service than an oil-firing system; and reduction of the labor to attend the boiler furnace while it is banked. The ease of starting an oil fire in comparison with a coal fire eliminates the necessity of maintaining a slow rate of combustion while it is not on active service. The boiler setting for an oil-fired furnace is kept hot by closing all dampers tightly to prevent infiltration and circulation of cold air through the flue-gas passages. This cannot be done with a coal fire on a stoker since provisions must be made to carry off the combustion gases of the slow fire carried on the grates, and a feed of air to this fire must be maintained.

33-13. Interconnections. When two power systems are in proximity, it is often economic to connect them so that energy may be interchanged. The economic value arises from three factors:

1. Each system individually operated must carry its own reserve capacity. By connection through adequate ties the reserve of one system may be sufficient to serve for both. This reduction in reserve capacity will allow taking the older inefficient units out of active service, thereby reducing total fuel consumption.

2. There often is a certain diversity in time between the peak demands of two systems, so that the peak load of the combined systems is less than the sum of the individual peaks. Because of this diversity and the additional reduction in reserve capacity just discussed the total active installed capacity requirement of two systems is reduced over that for individual operation. This will allow a postponement of additional new capacity with a growing load and effect important economies.

3. A redistribution of loading among the generating stations by dividing the load incrementally between two systems usually effects an immediate saving in fuel.

The method of determining the economic capacity of a proposed interconnection is rather lengthy, and only the major items are emphasized in the following example.

Example 33-10. Two power systems A and B, serving adjacent territories, are proposed to be interconnected for purposes of increasing the total economy of operation. The individual and combined daily load variations are tabulated in Table 33-21 and illustrated in Fig. 33-8.

For purposes of simplification assume that the given curves represent average as well as peak days.

The capacity of system *A* totals 280 mw and is comprised of four 50-mw, two 30-mw, and two 10-mw units. System *B* has a total capacity of 700 mw and consists of three 100-mw, four 50-mw, and ten 20-mw units. Each system has been designed for a two-unit installed reserve and operates on a one-unit spinning reserve. On this basis system *A* has a spinning reserve of 50 mw and an installed reserve of 100 mw. System *B* has a spinning reserve of 100 mw and an installed reserve of 200 mw.

TABLE 33-21. CHRONOLOGICAL LOAD CURVES OF SYSTEMS *A* AND *B* THAT ARE PROPOSED FOR INTERCONNECTION

Time	Load, megawatts		
	System *A*	System *B*	Combined loads
1 A.M.	70	160	230
2	50	140	190
3	40	120	160
4	30	120	150
5	30	120	150
6	40	130	170
7	60	160	220
8	110	180	290
9	160	200	360
10	180	200	380
11	180	200	380
12 M.	140	160	300
1 P.M.	170	200	370
2	180	200	380
3	180	200	380
4	180	250	430
5	155	370	525
6	135	500	635
7	120	450	570
8	110	380	490
9	105	330	435
10	100	280	380
11	95	230	325
12	90	180	270

Both systems are growing, and the peak load on each increases at an annual rate of about 10 per cent. This trend will apparently continue for some time as indicated by an investigation of the probable future energy requirements of the two regions being served. For continued isolated operation of each system this would require a minimum of 68 mw additional capacity yearly to maintain the established installed reserve policy. This would probably be provided in new units equaling in capacity the largest unit on each system, a sum of 150 mw.

The sum of the individual peak loads is 680 mw compared with a combined peak load of 635 mw, a difference of 45 mw due to diversity. As the capacity of the combined systems is 980 mw, the existing reserve for the combined systems is 345 mw.

For systems having more than about 10 units, experience indicates that accidental outage of two units simultaneously at the precise time of the peak load is a remote possibility (see Sec. 31-2). Hence, in merging two systems for joint operation, it is feasible to permit the reserve of the larger system to serve also as the total reserve of the combined system, provided that the interconnecting tie is of sufficient capacity.

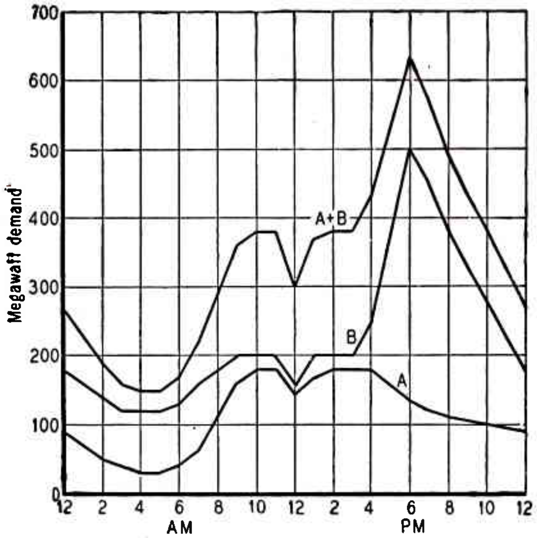


FIG. 33-8. Individual and combined daily load curves for systems *A* and *B*.

Since the largest units on the combined system are of 100-mw capacity, the installed reserve will be set at 200 mw compared with the 345 mw actually available. The difference of 145 mw is available for future growth, contrasting with the necessity of installing new capacity if isolated operation is pursued. Of course, an appreciable investment will be required to construct the interconnecting tie line and make the two cases of (1) isolated capacity expansion and (2) joint operation economically competitive.

Both should be studied to determine which course is the more desirable, not only in the light of immediate conditions but also with consideration of the probable effect on the future economy of operation by adopting either policy.

Transmission-line capacities and corresponding installed costs to interconnect systems *A* and *B* are being considered as in the following tabulation:

Kw	*Costs*
50,000	$1,400,000
100,000	2,200,000
150,000	2,800,000

The incremental-rate curve for each system is shown in Fig. 33-9, together with the incremental-rate curve for the combined systems.

The costs of operating the individual units on each system, either at no load or in

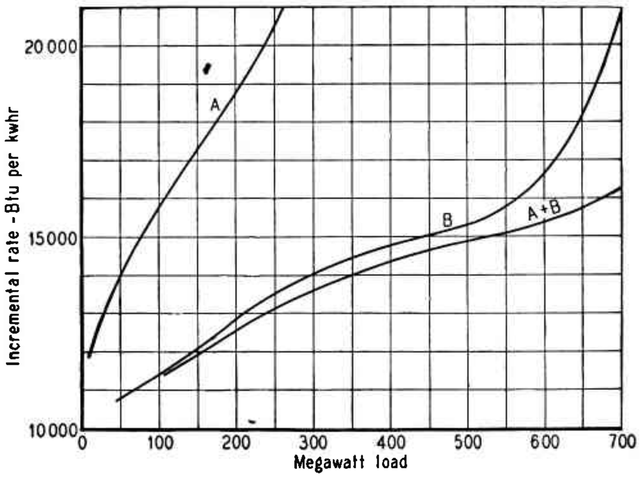


FIG. 33-9. Individual and combined incremental-rate curves for systems *A* and *B*.

a banked condition, including the cost of fuel, labor, and maintenance, are as in the following tabulation:

Units	*Cost per unit per year*
System *A*:	
50-mw	$150,000
30-mw	105,000
10-mw	50,000
System *B*:	
100-mw	200,000
50-mw	140,000
20-mw	80,000

An analysis of the comparative capacity requirements for each size of transmission line is given in Table 33-22. The load and capacity conditions must be studied at the times of the individual system peaks and the combined system peak to determine the units to be operated. Since the system *B* peak and the combined system peak occur at the same time, only this period and the period of the individual system *A* peak need be studied. To determine the savings effected in stand-by costs, the spinning capacities for isolated operation on *A* and *B* are also included.

In all cases, the objective is to eliminate all the high-cost stand-by capacity possible. This is found by reducing the total stand-by costs to a unit cost, as in the following tabulation:

Unit	*Cost per kilowatt-year*
System *A*:	
50-mw	$3.00
30-mw	3.50
10-mw	5.00
System *B*:	
100-mw	2.00
50-mw	2.80
20-mw	4.00

The order of unit preference on the basis of stand-by costs is

$$100 - B;\ 50 - B;\ 50 - A;\ 30 - A;\ 20 - B;\ \text{and}\ 10 - A$$

At each period (see Table 33-22) the minimum required spinning capacity is determined by adding to the system load the spinning reserve required. Considering the system *A* peak period for the 50-mw tie, the total combined spinning capacity required is 530 mw. By using the large units first, in preferred order of stand-by costs which roughly correspond with their average efficiency (not shown), the nearest combination provides 550 mw. With this placement of units on each system indicated as units operating (see table) it is then necessary to investigate whether each individual system has adequate protection. System *A* has 200 mw in spinning capacity; system *B* has a spinning capacity of 350 mw and a load of 250 mw, leaving a reserve of 100 mw. However, only 50 mw of this reserve is available to system *A* because of the tie capacity limitation. Then the total capacity available to system *A* is the sum of its spinning capacity and the tie capacity, or 250 mw. Its load at the time is 180 mw; therefore its reserve capacity is 70 mw, sufficient to protect it against the loss of the largest unit at its own end of the tie line.

System *B* has in spinning capacity 350 mw. The difference between the spinning capacity and load of system *A* is 20 mw, all of which is available to system *B*. Then the total capacity available to *B* is 370 mw. Subtracting its load of 250 mw, we find that it has an ample reserve of 120 mw.

For the time of the combined and system *B* peak a similar study shows that the actual spinning capacity must be 760 mw against an indicated minimum of 735 mw. At first glance it might seem that one 20-mw unit too many has been placed in service. Without this unit system *B* would have only 90-mw reserve as against the required 100 mw.

After all the critical periods have been analyzed, it develops that the time of the combined peak determines the amount of active capacity required. This is usually the case but is not always necessarily so. Comparing the no-load capacity costs, the 50-mw tie saves jointly 705,000 + 1,560,000 − 2,000,000 = $265,000 annually by eliminating the operation of one 30-mw and two 20-mw units. The 100-mw and 150-mw ties save a total of $345,000 annually in stand-by costs.

An estimate of the direct fuel saving that can be realized from each tie is calculated in Table 33-23, assuming that the transmission-line energy losses are negligible. The computations are carried out in the following manner. The combined system load at each hour (column 4) is the sum of the individual system loads in columns 2 and 3. The combined load is then divided incrementally between the two systems by Fig. 33-9 and tabulated in columns 5 and 6 as the energy generated by each system under combined operation.

The energy transmitted over the tie line (column 7) will then be evaluated by the difference of the hourly figures in either columns 2 and 5 or columns 3 and 6. The relative magnitudes determine the direction of transfer, in this case being from *B* to *A* at all hours. Inspection of all figures in column 7 shows that all magnitudes are less than the tie-line capacity of 150 mw.

Referring to the condition at 1 A.M., system *B* picks up 43 mw in generation, and *A* drops an equal amount, as compared with conditions of isolated operation. The decreased energy input to system *A* due to this transfer is evaluated by $\int_{70}^{27} IR\,dL$. Since the equations of the incremental-rate curves are not known, graphical means of evaluation must be employed. By assuming the incremental-rate curve to be a straight line from 27 to 70 mw the average height of the area under this portion of

TABLE 33-22. ANALYSIS OF COMPARATIVE CAPACITY AND RESERVE REQUIREMENTS AND STAND-BY COSTS

	Isolated operation		50-mw interconnection						100- and 150-mw interconnection					
			At time of system *A* peak			At time of system *B* peak[a]			At time of system *A* peak			At time of system *B* peak[a]		
System	*A*	*B*	*A*	*B*	*A* + *B*	*A*	*B*	*A* + *B*	*A*	*B*	*A* + *B*	*A*	*B*	*A* + *B*
Load	180	500	180	250	430	135	500	635	180	250	430	135	500	635
Spinning reserve	50	100	....	.....	100	....	......	100	....	.....	100	....	......	100
Minimum spinning capacity required	230	600	....	.....	530	....	......	735	....	.....	530	....	......	735
Actual spinning capacity	230	600	200	350	550	200	560	760	200	350	550	200	540	740
Tie capacity	....	......	50	20	.....	50	50	.....	100	20	.....	40	65	
Total capacity available to systems	230	600	250	370	550	250	610	760	300	370	550	240	605	740
Less load	180	500	180	250	430	135	500	635	180	250	430	135	500	635
Actual spinning reserve	50	100	70	120	120	115	110	125	120	120	120	105	105	105
Units operating	4–50 1–30	3–100 4– 50 5– 20	4–50	3–100 1– 50	.....	4–50	3–100 4– 50 3– 20	.....	4–50	3–100 1– 50	.....	4–50	3–100 4– 50 2– 20	
No-load capacity cost, $1,000	705	1,560	600	740	1,340	600	1,400	2,000	600	740	1,340	600	1,320	1,920

[a] **The combined system peak occurs at the same time as system *B* peak.**

TABLE 33-23. CALCULATION OF DAILY AND ANNUAL DIRECT FUEL SAVINGS EFFECTED BY EACH OF THREE PROPOSED TIE LINES

INTERCONNECTING SYSTEMS *A* AND *B*

Time	Mw load			Mw generation		150-mw transmission line					100-mw transmission line					50-mw transmission line				
	A	*B*	*A* + *B*	*A*	*B*	Mw transmission, *B* to *A*	Decrement rate system *A* Btu per kwhr	Increment rate system *B* Btu per kwhr	Differential increment rate *A* over *B*	Fuel saving, 10^6 Btu per hr	Mw transmission, *B* to *A*	Decrement rate system *A* Btu per kwhr	Increment rate system *B* Btu per kwhr	Differential increment rate *A* over *B*	Fuel saving, 10^6 Btu per hr	Mw transmission *B* to *A*	Decrement rate system *A* Btu per kwhr	Increment rate system *B* Btu per kwhr	Differential increment rate *A* over *B*	Fuel saving, 10^6 Btu per hr
(1)	(2)	(3)	(4)	(5)	(6)	(7) = (2) − (5) = (6) − (3)	(8)	(9)	(10) = (8) − (9)	(11) = (7) × (10)	(12)	(13)	(14)	(15)	(16)	(17)	(18)	(19)	(20)	(21)
1 A.M.	70	160	230	27	203	43	13,820	12,580	1,240	53.3	43	......	......	......	53.3	43	......	......	......	53.3
2	50	140	190	18	172	32	13,180	12,200	980	31.4	32	......	......	......	31.4	32	......	......	......	31.4
3	40	120	160	13	147	27	12,800	11,880	920	24.8	27	......	......	......	24.8	27	......	......	......	24.8
4	30	120	150	11	139	19	12,540	11,830	710	13.5	19	......	......	......	13.5	19	......	......	......	13.5
5	30	120	150	11	139	19	12,540	11,830	710	13.5	19	......	......	......	13.5	19	......	......	......	13.5
6	40	130	170	13	157	27	12,800	12,020	780	21.1	27	......	......	......	21.1	27	......	......	......	21.1
7	60	160	220	24	196	36	13,560	12,530	1,030	37.1	36	......	......	......	37.1	36	......	......	......	37.1
8	110	180	290	40	250	70	14,780	13,040	1,740	121.8	70	......	......	......	121.8	50	15,220	12,920	2,300	115.0
9	160	200	360	55	305	105	15,850	13,470	2,380	249.9	100	15,950	13,450	2,500	250.0	50	16,790	13,200	3,590	179.5
10	180	200	380	58	322	122	16,210	13,550	2,660	324.5	100	16,600	13,450	3,150	315.0	50	17,390	13,200	4,190	209.5
11	180	200	380	58	322	122	16,210	13,550	2,660	324.5	100	16,600	13,450	3,150	315.0	50	17,390	13,200	4,190	209.5
12 M.	140	160	300	45	255	95	15,420	12,920	2,500	237.5	95	......	......	......	237.5	50	16,190	12,630	3,560	178.0
1 P.M.	170	200	370	57	313	113	16,040	13,510	2,530	285.9	100	16,280	13,450	2,830	283.0	50	17,100	13,200	3,900	195.0
2	180	200	380	58	322	122	16,210	13,550	2,660	324.5	100	16,600	13,450	3,150	315.0	50	17,390	13,200	4,190	209.5
3	180	200	380	58	322	122	16,210	13,550	2,660	324.5	100	16,600	13,450	3,150	315.0	50	17,390	13,200	4,190	209.5
4	180	250	430	65	365	115	16,330	14,030	2,300	264.5	100	16,600	13,980	2,620	262.0	50	17,390	13,780	3,610	180.5
5	155	370	525	85	440	70	16,330	14,780	1,550	108.5	70	......	......	......	108.5	50	16,650	14,740	1,910	95.5
6	135	500	635	98	537	37	16,230	15,460	770	28.5	37	......	......	......	28.5	37	......	......	......	28.5
7	120	450	570	95	475	25	15,970	15,110	860	21.5	25	......	......	......	21.5	25	......	......	......	21.5
8	110	380	490	75	415	35	15,470	14,750	720	25.2	35	......	......	......	25.2	35	......	......	......	25.2
9	105	330	435	65	370	40	15,220	14,430	790	31.6	40	......	......	......	31.6	40	......	......	......	31.6
10	100	280	380	58	322	42	15,000	14,030	970	40.7	42	......	......	......	40.7	42	......	......	......	40.7
11	95	230	325	48	277	47	14,770	13,560	1,210	56.9	47	......	......	......	56.9	47	......	......	......	56.9
12	90	180	270	37	233	53	14,400	12,940	1,460	77.4	53	......	......	......	77.4	50	14,460	12,920	1,540	77.0
Total	2,710	5,460	8,170	1,172	6,998	1,538	......	......	.....	3,042.4	1,417	......	......	......	2,999.3	979	......	......	......	2,257.6
Daily fuel saving at 15 cents per million Btu										$456					$450					$338
Annual fuel saving										$166,000					$164,000					$123,000

the curve will be (12,920 + 14,720)/2 = 13,820 Btu per kwhr, i.e., the average of the incremental rates at 27- and 70-mw loads. The error of the foregoing assumption is entirely negligible, though theoretically the average height of the area is slightly greater than this average since the curve is concave downward. Then each kilowatt in this block of 43 mw requires 13,820 Btu if produced on system *A*. Since this block is being dropped, the product 13,820 × 43,000 = 595 × 10^6 Btu per hr represents the decreased energy input to system *A* when interconnected with system *B*. To indicate the direction of application, the increment rate of 13,820 Btu per kwhr may be termed the "decrement" rate.

Now system *B* picks up this block of 43 mw by increasing its generation from 160 to 203 mw. The average incremental rate at which it is picked up is (12,240 + 12,920)/2 = 12,580 Btu per kwhr. Comparing the increment and decrement rates, we find that a saving of 13,820 − 12,580 = 1240 Btu per kwhr for the 43-mw block is effected by the energy transfer over the tie line, or a total saving of 43,000 × 1,240 = 53.3 × 10^6 Btu per hr.

For the ties of lesser capacity all of the transfer indicated as desirable on an incremental basis cannot be effected. For example, at 3:00 P.M. though the ideal transfer is 122 mw, with ties of 100- and 50-mw capacity the maximum energy that can be transferred is limited to the capacity of the transmission line, and the economies are calculated on the basis of the smaller transfers.

To sum up the hourly savings for each tie, we find the total savings decrease with decrease in capacity. The difference in savings effected by the 150- and 100-mw ties is negligible but much more marked for the 100- and 50-mw ties.

The daily average energy savings per kilowatthour transferred compare as in the following tabulation:

Megawatts	*Btu per kwhr*
150	1980
100	2115
50	2305

In this example for purposes of simplification the load curves are assumed to be duplicated 365 days a year. Then the annual savings are 365 times the daily saving.

TABLE 33-24. COMPARISON OF INTERCONNECTION ECONOMIES OF THREE PROPOSED TIE LINES

1. Tie-line capacity	50 mw	100 mw	150 mw
2. Joint fuel savings	$ 123,000	$ 164,000	$ 166,000
3. Savings by elimination of stand-by capacity:			
System *A*	105,000	105,000	105,000
System *B*	160,000	240,000	240,000
4. Total joint savings	$ 388,000	$ 509,000	$ 511,000
5. Joint tie-line investment	1,400,000	2,200,000	2,800,000
6. Rate of return available for fixed charges, per cent	27.7	23.1	18.2
7. Differential joint saving	$121,000		$ 2,000
8. Differential tie-line investment	800,000		600,000
9. Rate of return available for fixed charges on increment investment, per cent	15.1		0.33

With the joint fuel savings known, as well as the joint reduction in operating capacity and the required investments, the comparative economy of the three schemes can be calculated as in Table 33-24.

On the assumption that both systems are owned by the same management, the relative financial merit of the alternate proposals could be decided by a study of Table 33-24. If the policy of the company required an 18 per cent return or better on the investment, it might seem at first glance that all schemes would amply satisfy this requirement, as shown by line 6 of the table. However, the true comparison is given by line 9; the additional investment of $800,000 required by the 100-mw tie as compared with the 50-mw tie gives an increase in savings of $121,000 annually or a return of 15.1 per cent on this additional differential investment. Since this does not meet the minimum return required, the 50-mw tie would be the choice. The 150-mw tie is obviously uneconomical, owing to the differential investment return of only 0.33 per cent.

The tie-line proposal must be compared with new capacity installations as mentioned previously. Each system would probably install units at least equal in capacity to the largest existing units. The study of such a proposal may readily show the results assumed in Table 33-25.

TABLE 33-25. CALCULATION OF ECONOMY OF INCREASING SYSTEM CAPACITY BY INSTALLATION OF NEW UNITS, FOR FIRST YEAR OF OPERATION

1. System	*A*	*B*	Combined
2. New capacity	50 mw	100 mw	150 mw
3. Investment in new capacity	$5,000,000	$9,000,000	$14,000,000
4. Annual fuel saving	300,000	500,000	800,000
5. Saving in stand-by elimination	105,000	320,000	425,000
6. Total annual savings	$ 405,000	$ 820,000	$ 1,225,000
7. Rate of return available for fixed charges, per cent	8.1	9.1	8.8

By comparing the combined capacity extension and the 50-mw tie proposals the possibility of achieving superior economies by such interconnections is quite apparent. However, the reader should understand that all interconnection proposals are not necessarily feasible. This example was chosen with conditions that were all favorable toward interconnection. This may not be the case in actual circumstances.

In this example the systems are about 100 miles apart, and the land necessary for the transmission line has been assumed to be no more expensive than ordinary farm lands. Several significant comparisons can be made of the two alternate courses of action. The tie line avoids a combined immediate investment of $14,000,000 for new capacity less the cost of the line, a net amount of $11,200,000, in addition to making available 145 mw of present capacity to absorb future growth, which in turn avoids future investment. The tie line makes available an additional profit out of savings which can go partly toward reducing energy rates.

If each system continued operating independently, the fixed costs would not yield savings that would in any way allow an extra margin of profit. Of course, the added fixed charges caused by the addition of capacity will generally be covered by the additional revenues received from the increased load in subsequent years and will allow the company the legal rate of return or somewhat less in the normal course of growth. But the trend in rate reductions is necessarily slower owing to the higher investment required for isolated operation.

The economic solution of an interconnection problem as demonstrated by the above discussion can be resolved into four major steps:

1. Assemble the chronological load data for individual systems, and determine the combined hourly demands.
2. Study the distribution of capacity in each system to achieve the maximum fuel economy and protect against accidental outage in any part of the combined systems at all times.
3. Compute the annual fuel savings resulting from energy transfers over ties of various capacities.
4. Compare the net savings with the corresponding investments to determine the rate of return.

PROBLEMS

33-1. An industrial organization studying the relative desirability of two diesel engines proposed for installation has estimated that the annual operating costs for engine A is \$40,000 and for engine B is \$35,000. What is the added justifiable investment for engine B if the cost of money use is 8 per cent of investment, taxes are 4 per cent of 80 per cent of the first cost, insurance is 0.3 per cent of investment, and the annual depreciation charge has been set at 5 per cent of the total investment?

33-2. A village is considering the installation of a 1,500-kw municipal plant composed of two diesel engines. Three different plans of installation are proposed as in Table P-1.

TABLE P-1

Plant	Total installed cost per kw	Input-output curve for one engine
A	\$200	$I = 2{,}500{,}000 + 5{,}000L + 0.006L^3$
B	230	$I = 1{,}900{,}000 + 4{,}000L + 0.006L^3$
C	250	$I = 1{,}400{,}000 + 3{,}000L + 6L^2 - 0.0005L^3$

I is in Btu per hour, and L is in kilowatts.

The annual load requirements are as shown in Table P-2.

TABLE P-2

Load, kw	*Hr at load*
1,400	1,000
1,000	4,000
500	1,000
100	2,760

No reserve capacity is being provided, and operation will be guided solely by fuel economy. Oil is available at \$2.10 per barrel of 360 lb with a heating value of 19,000 Btu per lb. If the fixed-charge rate is 11 per cent, which is the economic plant?

33-3. Find the economic plant for the conditions of Prob. 33-2 if the load requirements are as shown in Table P-3.

TABLE P-3

Load, kw	*Hr at load*
1,400	500
800	7,000
100	1,260

33-4. Find the economic plant for the conditions of Prob. 33-2 if the fixed-charge rate is 7 per cent.

33-5. Find the economic plant for the conditions of Prob. 33-2 if the installed costs are as in Table P-4.

TABLE P-4

Plant	*Unit cost per kw*
A	\$200
B	210
C	220

33-6. Find the economic plant for the conditions of Prob. 33-2 if the fuel price is 45 cents per million Btu.

33-7. Find the economic plant for the conditions of Prob. 33-2 if the fuel price is 45 cents per million Btu and the fixed-charge rate is 7 per cent.

33-8. A high-pressure steam plant and a diesel-engine plant are being compared, each having a capacity of 1,000 kw. The load to be served has the variations given in Table P-5.

TABLE P-5

Load, kw	*Hr at load*
1,000	100
800	1,000
500	6,000
200	1,660

The performance characteristics of the plants are

Steam: $I = 1{,}300{,}000 + 6{,}000L + L^2 + 0.004L^3$
Diesel: $I = 1{,}100{,}000 + 6{,}000L + 2.6L^2 - 0.0007L^3$

where I is in Btu per hour and L is in kilowatts.

The fuel costs are as follows: for the steam plant, coal having a calorific value of 14,500 Btu per lb costs \$6 per short ton; for the diesel plant, oil having a calorific value of 17,000 Btu per lb costs \$2.10 per barrel of 360 lb.

The steam plant will require a crew of eight men at an average annual salary of \$3,500. The diesel plant requires a crew of five men at the same salary rate. Maintenance expenditures for the steam plant are estimated at \$2,000 per year; for the diesel plant \$1,500 annually.

The installed costs are estimated as follows: steam plant at \$220 per kilowatt and diesel plant at \$200 per kilowatt. Which plant is the more economical if the fixed-charge rate is 15 per cent?

33-9. The data of Table P-6 have been assembled in comparing two proposed installations of 1,000-kw capacity each.

TABLE P-6

	Steam plant	Diesel plant
Fuel cost, cents per million Btu	25.0	30.0
Annual maintenance cost	\$ 2,000	\$ 2,200
Annual labor cost	18,000	16,000
Annual supplies cost	5,000	4,000
Unit installation cost, per kw	300	330

The load has the characteristics of Table P-7.

TABLE P-7

Load, kw	*Hr at load*
1,000	100
800	1,000
500	6,000
200	1,660

Plant performance characteristics are

Steam: $I = 1{,}300{,}000 + 6{,}000L + L^2 + 0.004L^3$
Diesel: $I = 1{,}100{,}000 + 6{,}000L + 2.6L^2 - 0.0007L^3$

where I is in Btu per hour and L is in kilowatts.

At a fixed-charge rate of 11 per cent which is the economic plant?

33-10. Find the oil unit fuel cost that would make the two proposals of Prob. 33-9 equally economical.

33-11. Which plant will be economic in Prob. 33-9 if the load demands of Table P-8 were to be supplied?

TABLE P-8

Load, kw	*Hr at load*
1,000	4,000
800	4,000
500	760

33-12. For a given installation a turbine manufacturer offers two alternate turbines of 1,000 kw capacity. Turbine A is valved to develop maximum efficiency at half load, turbine B to develop maximum efficiency near full load. The performance characteristics of the two turbines are

A: $I = 1{,}300{,}000 + 6{,}000L + L^2 + 0.004L^3$
B: $I = 1{,}300{,}000 + 7{,}000L + 1.4L^2 + 0.0004L^3$

Which turbine should be selected for the load conditions given in Table P-9?

TABLE P-9

Load, kw	*Hr at load*
1,000	200
800	2,000
500	4,000
200	2,560

The installed costs of the two turbines are the same.

33-13. An industrial organization is considering the installation of a 3,000-kw diesel engine plant. Reserve capacity is not considered necessary, and a policy requiring equal-sized units has been established. To arrive at proper proportioning of unit size, the plans of Table P-10 are being studied.

TABLE P-10

No. units	Unit capacity, kw	Installed cost per kw	Total housing cost
3	1,000	\$250	\$20,000
2	1,500	230	15,000
1	3,000	220	10,000

The performance characteristics of the engines are

1,000 kw: $I = 1{,}100{,}000 + 8{,}000L + 2.6L^2 - 0.0007L^3$
1,500 kw: $I = 2{,}000{,}000 + 7{,}500L + 1.4L^2 - 0.0002L^3$
3,000 kw: $I = 3{,}700{,}000 + 7{,}000L + 0.8L^2 - 0.00007L^3$

where I is in Btu per hour and L is in kilowatts.

Maintenance cost per unit is estimated at \$1,000 annually regardless of size. The labor costs for the one- and two-unit plants are equal, but two extra men will be required at a salary of \$2,000 annually for the three-unit plant. Fuel costs 22 cents per million Btu. The expected load demand is shown in Table P-11.

TABLE P-11

Load, kw	*Hr at load*
3,000	1,000
2,000	2,000
1,000	2,000
500	3,760

Which is the economic plan at a fixed-charge rate of 18 per cent?

33-14. A town desires the installation of a 3,000-kw diesel engine plant to supply electrical energy. The requirements are not considered critical enough to need reserve capacity in the plant. The load to be generated annually is shown in Table P-12.

TABLE P-12

Load, kw	*Hr at load*
3,000	200
2,000	1,000
1,000	2,000
500	5,560

To determine the optimum size of plant the following proposed engines, costs, and performance data are being studied:

TABLE P-13

No. engines	Engine capacity, kw	Unit installed cost per kw
1	3,000	$250
2	1,500	260
3	1,000	275

3,000 kw: $I = 6{,}100{,}000 + 5{,}000L + 1.2L^2 - 0.00007L^3$
1,500 kw: $I = 3{,}400{,}000 + 5{,}000L + 2.3L^2 - 0.0003L^3$
1,000 kw: $I = 2{,}100{,}000 + 5{,}000L + 3.6L^2 - 0.0007L^3$

where I is in Btu per hour and L is in kilowatts.

All other annual costs are expected to be equal for all plans. Find the optimum plant at a fixed-charge rate of 10 per cent and a fuel cost of 30 cents per million Btu.

33-15. An electrical supply system is experiencing a uniform rate of load growth amounting to 2 mw per year increase in magnitude of the peak load. The annual load factor and the shape of the annual load-duration curve remain constant during the growth as described in Table P-14.

TABLE P-14

Load, per cent of peak load	*Time, per cent of total time at load*
100	5
80	10
60	50
40	20
20	15

For a given year the peak load was 20 mw, and five 5-mw units supplied the load. A one-unit operating reserve was always maintained.

a. Plot the load-duration curve for the year, and indicate thereon the generation of each unit, labeling the least efficient *A*.

b. Plot the load-duration curve for a year 10 years afterward, and indicate thereon the generation of each unit, assuming that the newer units are most efficient, that a one-unit operating reserve has been maintained, and that each new unit is of 5 mw capacity.

33-16. The average daily-load curve of a 100-mw central station is shown in Table P-15.

TABLE P-15

Time	Average hourly output, mwhr	Time	Average hourly output, mwhr
12 P.M.– 7 A.M.	10	6 P.M.– 7 P.M.	60
7 A.M.– 8 A.M.	30	7 P.M.– 8 P.M.	50
8 A.M.–12 A.M.	50	8 P.M.– 9 P.M.	40
12 A.M.– 1 P.M.	40	9 P.M.–10 P.M.	30
1 P.M.– 4 P.M.	50	10 P.M.–11 P.M.	20
4 P.M.– 5 P.M.	60	11 P.M.–12 P.M.	10
5 P.M.– 6 P.M.	70		

The performance of the station is closely defined by

$$I = 10^6 \times (76 + 7L + 0.08L^2 + 0.00006L^3)$$

where I is in Btu per hour and L is in megawatts.

Plot the heat-rate and incremental-rate curves of the station and the chronological load curve.

The maintenance costs of the plant are a direct function of the load and are 15 per cent of the fuel cost. The unit fuel cost is 20 cents per million Btu.

Find

a. The daily fuel and maintenance costs of supplying the total load.

b. The average unit fuel and maintenance costs per kilowatthour for the daily load.

c. The average unit fuel and maintenance costs of adding a constant load of 10 mw for 24 hr of the day. Do this in two ways, by finding the increased input from the *I-L* curve and from the *IR-L* curve.

d. The same as (*c*), but with a constant load of 20 mw.

e. The same as (*c*), but with a constant load of 30 mw.

f. The unit cost for loads of 10, 20, and 30 mw added between 11 P.M. and 7 A.M.

g. The unit cost for loads of 10, 20, and 30 mw added between 8 A.M. and 4 P.M.

h. The unit cost for loads of 10, 20, and 30 mw added between 5 P.M. and 6 P.M.

33-17. A large industrial plant has a 50-mw steam power station, the performance characteristic of which is

$$I = 10^6 \times (44 + 12L + 0.002L^3)$$

where I is in Btu per hour and L is in megawatts. The plant is being expanded and an additional 10-mw unit is to be built to take care of the expected loads shown in Table P-16.

TABLE P-16

Load, mw	*Hr at load*
50	2,000
30	3,000
10	3,760

Two different units are under study: a low-pressure unit *A* costing $70 per kilowatt and a high-pressure unit B costing \$100 per kilowatt. The performance characteristics are

$$A: \quad I = 10^6 \times (3 + 11L + 0.03L^3)$$
$$B: \quad I = 10^6 \times (20 + 7L + 0.01L^3)$$

All other costs are estimated to be the same for both units.

a. Which unit should be installed if the fuel cost is 20 cents per million Btu and the fixed charge rate is 15 per cent? Assume that the existing system is kept hot when generation is reduced to zero and has an availability of 85 per cent.

b. Find the fuel cost saved by installing the new units.

c. Find the average unit fuel cost of the generation dropped by the existing plant when operating with each of the proposed units.

33-18. An industrial plant with a 2,000-kw maximum demand for power has two alternate methods by which it may supply its energy requirements: (*a*) install a diesel electric plant or (*b*) install a low-head hydraulic plant. The annual energy requirements are given in Table P-17.

TABLE P-17

Load, kw	*Hr at load*
2,000	500
1,600	3,000
400	4,500
0	760

No reserve capacity is required, and during zero load no engine is required to be kept in operation.

The optimum diesel plant has been found to be two 1,000-kw engine units having an installed cost of \$220 per kilowatt. The input-output characteristic of one engine is given by

$$I = 10^6 \times (2 + 8L + 2L^2)$$

where I is in Btu per hour and L is in megawatts.

The diesels will use oil having a heating value of 19,000 Btu per lb and costing \$1.20 per barrel of 350 lb. They will also require 0.1 qt of lubricating oil per engine-hour of operation at 20 cents per quart. Annual maintenance costs are estimated at 15 per cent of annual fuel cost. A crew of four men will be required for operation, an engineer at \$5,000 per year and three operators at \$3,500 per year.

A hydraulic plant of equal capacity can be installed at a cost of \$300 per kilowatt. The annual license fee for riparian rights amounts to 1 per cent of the investment. The operating crew requires the same pay-roll expenditures as for the diesel plant. Annual maintenance is estimated at \$1,500. The hydraulic plant can meet the load demand at all times.

a. If the fixed charge rate is 15 per cent, which plant should be selected?

b. Find the unit installation cost of the hydraulic plant that would place it on a parity with the diesel plant.

c. Find the fixed-charge rate that would place both plants on a parity.

33-19. A system of 200-mw capacity requires additional capacity of 20 mw. The choice has been narrowed to the installation of a high-pressure steam plant or a hydraulic plant. The steam plant can be installed at a cost of \$100 per kilowatt, and its performance is defined by

$$I = 10^6 \times (32 + 7L + 0.004L^3)$$

The performance of the existing 200-mw system is closely defined by

$$I = 10^6 \times (140 + 10L + 0.0005L^3)$$

where I is in Btu per hour and L is in megawatts.

The expected loads on the total system are given in Table P-18.

TABLE P-18

Load, mw	*Hr at load*
200	100
120	5,000
80	3,660

The hydraulic station can be installed at a cost of \$200 per kilowatt, and the available river flow and storage provisions will limit the annual capacity factor to 70 per cent. The reduction in fuel cost in the existing system will be accomplished at the same differential unit fuel cost as for the alternate steam-plant proposal. The annual maintenance cost of the steam plant is estimated at 20 per cent of the fuel cost and the hydraulic maintenance estimated at 30 per cent of the steam-plant maintenance. Annual labor cost for the steam plant is \$30,000 and for the hydraulic plant \$10,000. Fuel costs 18 cents per million Btu. The fixed-charge rate for the steam plant is 12 per cent, for the hydraulic plant 10 per cent. Which plant should be installed? Assume availabilities of 90 per cent.

33-20. A municipal plant consisting of two 1,000-kw diesel engines needs capacity protection for an increasing load. Interconnection with an industrial plant containing two 500-kw diesels is being considered to provide the additional capacity. The municipal plant requires capacity protection for all loads in excess of 1,000 kw; the industrial plant does not need this protection.

The average daily-load curve of the municipal plant and the industrial plant are shown in Table P-19.

TABLE P-19

Municipal plant		Industrial plant	
Time	Load, kw	Time	Load, kw
12 P.M.– 8 A.M.	400	12 P.M.– 8 A.M.	100
8 A.M.– 5 P.M.	1,500	8 A.M.– 4 P.M.	800
5 P.M.– 8 P.M.	2,000	4 P.M.–12 P.M.	100
8 P.M.–11 P.M.	1,500		
11 P.M.–12 P.M.	400		

The performance characteristic of each of the 1,000-kw diesels is

$$I = 3{,}000{,}000 + 5{,}000L + 0.002L^3$$

The performance characteristic of each of the 500-kw diesels is

$$I = 1{,}600{,}000 + 7{,}000L + 0.007L^3$$

where I is in Btu per hour and L is in kilowatts.

a. If fuel costs 25 cents per million Btu, what joint daily fuel savings will be realized?

b. If the cost of effecting the interconnection is $1,000, does this scheme appear attractive compared with installing a new 1,000-kw engine at $150 per kilowatt and a fixed-charge rate of 10 per cent?

33-21. A 10,000-kw steam station is to be interconnected with a 1,000-kw diesel-engine plant. The daily loads on the steam station and the diesel station are shown in Table P-20. The performance characteristics of the stations are:

TABLE P-20

Steam plant		Diesel plant	
Time	Load, kw	Time	Load, kw
12 P.M.– 8 A.M.	1,000	12 P.M.– 8 A.M.	200
8 A.M.– 5 P.M.	5,000	8 A.M.– 4 P.M.	800
5 P.M.– 8 P.M.	10,000	4 P.M.–12 P.M.	200
8 P.M.–11 P.M.	5,000		
11 P.M.–12 P.M.	1,000		

Diesel: $I = 10^6 \times (3.0 + 5L + 2L^3)$
Steam: $I = 10^6 \times (24 + 8L + 0.09L^3)$

where I is in Btu per hour and L is in megawatts.

If the fuel cost for each plant is 20 cents per million Btu, what is the daily fuel saving?

33-22. A power-generation station is to be installed to handle a load of 250,000 kw. The daily load variation is as follows:

250,000 kw 10% of time
200,000 kw 30% of time
150,000 kw 30% of time
100,000 kw 30% of time

The generation will be either steam or hydro or a combination of the two types.

The only available hydro site that can be developed has an average flow of 2,000 cfs. By construction of a dam the water can be stored and used as needed. Head is 300 ft. The turbine efficiency for this site would be 85 per cent, and the generator efficiency would be 97.5 per cent. The capital charge (to cover taxes, insurance, depreciation, and return on investment) would be 10 per cent. Cost of operation and maintenance of the station would be $10 per hour for a one-unit station, $15 for two units, $20 for three units, $25 for four units, and $30 for five units.

Steam units having a net output of 50,000 kw can be installed at a cost of $100 per kilowatt for the first unit and $175 per kilowatt for the second, third, fourth, and fifth units. The steam unit capital charge is higher—12½ per cent.

Fuel would be coal at $7.75 per ton and 13,400 Btu per lb. Assume that the station net heat rate (from coal pile to high-voltage system) is 10,500 Btu per kwhr.

Cost of maintenance and operation of a steam unit is dependent on the station load factor. Assume operation and maintenance cost to vary as in Table P-21.

TABLE P-21

Load factor	Operation and maintenance, mills per kwhr
1.00	0.60
0.80	0.65
0.60	0.70
0.40	0.70

$$\text{Station load factor} = \frac{\text{kwhr output}}{\text{rated kw} \times \text{no. of hr}}$$

For simplification of the problem, assume that only 50,000-kw steam and hydro units will be considered and that the 250,000-kw load and generation are an independent system.

a. How much can be spent to develop the hydro site for a 50,000-kw unit? Give the answer in dollars per kilowatt, based on the difference in cost of a five- vs. four-unit system.

b. How much can be spent to install a second 50,000-kw unit?

c. How much for the third, fourth, and fifth hydro units?

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CHAPTER 34

ENERGY RATES

34-1. Energy-rate Objectives. Wherever energy is used, some method must be devised to distribute equitably the cost of supplying it among the various classifications of use. This problem presents itself regardless of whether the energy is used by the supplier, as in industrial enterprises owning private plants, or whether the energy is sold, as by a public electric utility. All forms of energy rates must cover the following items: (1) recovery of the cost of capital investment in generating equipment, transmission and distribution systems, and other forms of operating equipment; (2) recovery of the cost of operation, supplies, and maintenance of the equipment; (3) recovery of the cost of metering equipment, billing and collection costs, and miscellaneous services; and (4) a satisfactory net return on the total capital investment.

Although the determination of each cost item is relatively simple, the allocation of these items among the various classes of use to arrive at equitable charges is rather complex for the public utility and the large industrial plant and requires considerable engineering judgment and arbitrary determinations. These charges are expressed in terms of an *energy rate form*. The various forms used can be classified according to seven principal types, as discussed in the following sections. A short résumé of some of the problems of transforming total annual costs into these rate forms concludes this chapter.

34-2. General Rate Form. A large variety of rate forms are in actual use. However, they are all derivatives of the following general equation based on the recovery of the cost,

$$y = dx + ez + c$$

where y = total amount of bill for period during which energy was consumed, dollars
d = unit charge for metered or allowed maximum demand during billing period, dollars per kw
x = maximum demand during billing period, kw
e = unit charge for energy, dollars per kwhr
z = total energy consumed during period, kwhr
c = a constant charge made each billing period

The total charge is made up of three elements: one depending upon the maximum demand for energy, the second depending upon the total energy required, and the third being a constant figure. The first element, or demand charge, recovers a part of those investment charges that vary with the capacity of the generating and distributing equipment required; it also recovers a part of the operating costs that are independent of the total energy supplied but vary with the maximum rate at which energy may have to be supplied. The second element, or energy charge, recovers the remaining part of the investment charges of the generating and distributing equipment (the reason for this will be evident in the discussion closing this chapter) and those elements of operating costs which vary with the total energy produced. The third element, or customer charge, recovers the investment charges and operating expenses that are independent of demand or total energy. Thus a consumer that remains connected to the lines of a supplier but uses no energy incurs expenses for the supplier that must be paid for.

34-3. Flat Demand Rate. This is probably the first of the two early systems of charging applied to the field of the electric utility about the year 1882. It was based upon the total number of lamps installed and a fixed number of hours of use yearly. Thus the total demand and the energy consumption were fixed. The charge could then be distributed in even monthly or weekly payments. As all customers were of a like class, the rate could be expressed directly as a certain price per lamp or unit of installed capacity.

As the application of electricity broadened, the flat demand rate became restricted to uses such as sign lighting, signal systems, street lighting, and other loads where the energy consumption in point of quantity and time could be readily predicted.

By eliminating the metering equipment and the attendant reading and accounting costs the total charge can be materially reduced with the probability of attracting additional business of this type.

The flat demand rate can be expressed in the form

$$y = dx$$

Under this rate the consumer can theoretically use any amount of energy up to that consumed by his apparatus running at 100 per cent use factor, i.e., continuously at full load. With increased energy usage the unit energy cost becomes progressively lower since the total cost remains constant. The variation in total cost and unit cost is shown graphically in Fig. 34-1. Most rate forms can be plotted against the three coordinates of energy, demand, and total cost, thereby describing a geometric surface of definite characteristics. The surface of the flat demand rate has a definite limit when interpreted for practical usage. This limit is the plane perpendicular to the energy-demand plane, *o-a-b*. This plane is the locus of the maximum energy that can be consumed with a given demand within a given period and is termed the "100 per cent use factor" plane.

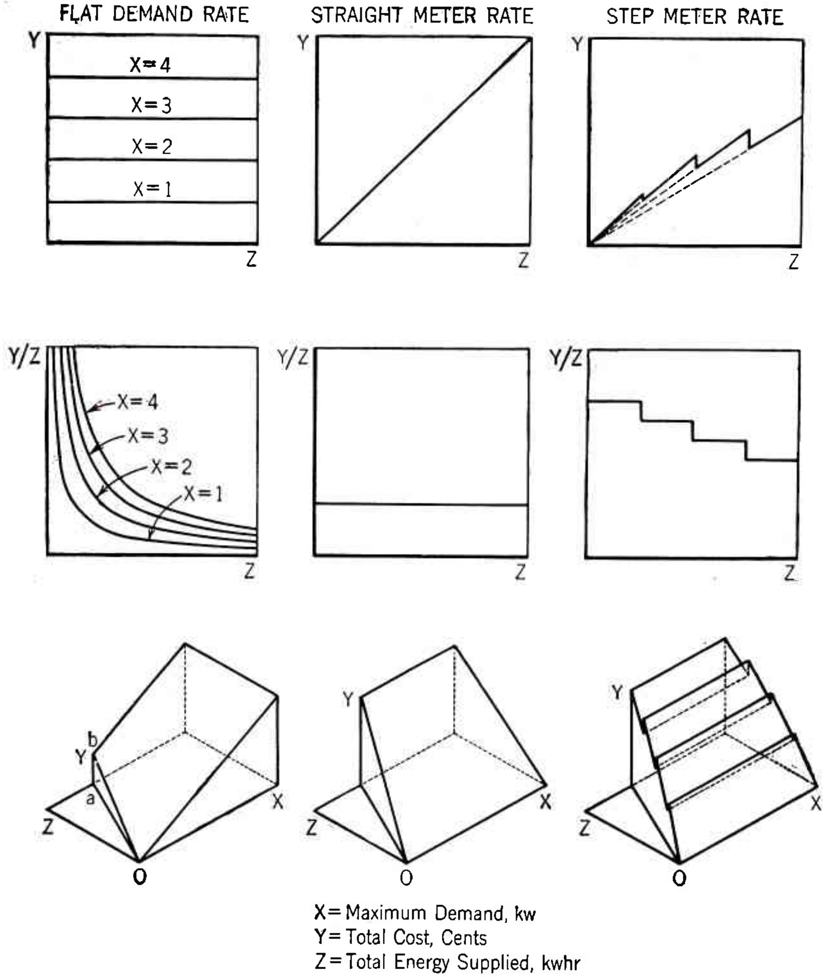
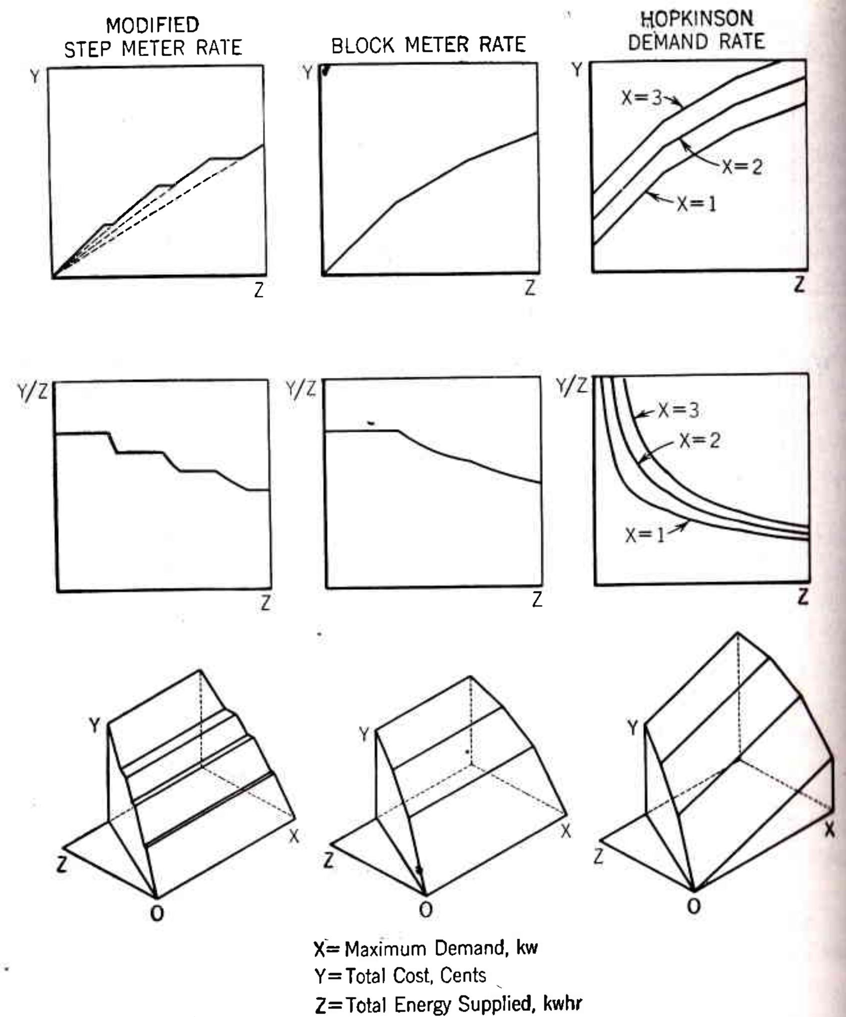


FIG. 34-1. Basic forms of electrical-energy service rates.

34-4. Straight Meter Rate. This rate is the second of the two forms of early systems of charging. It was instituted when a practical device for metering was devised and reduces to the form

$$y = ez$$

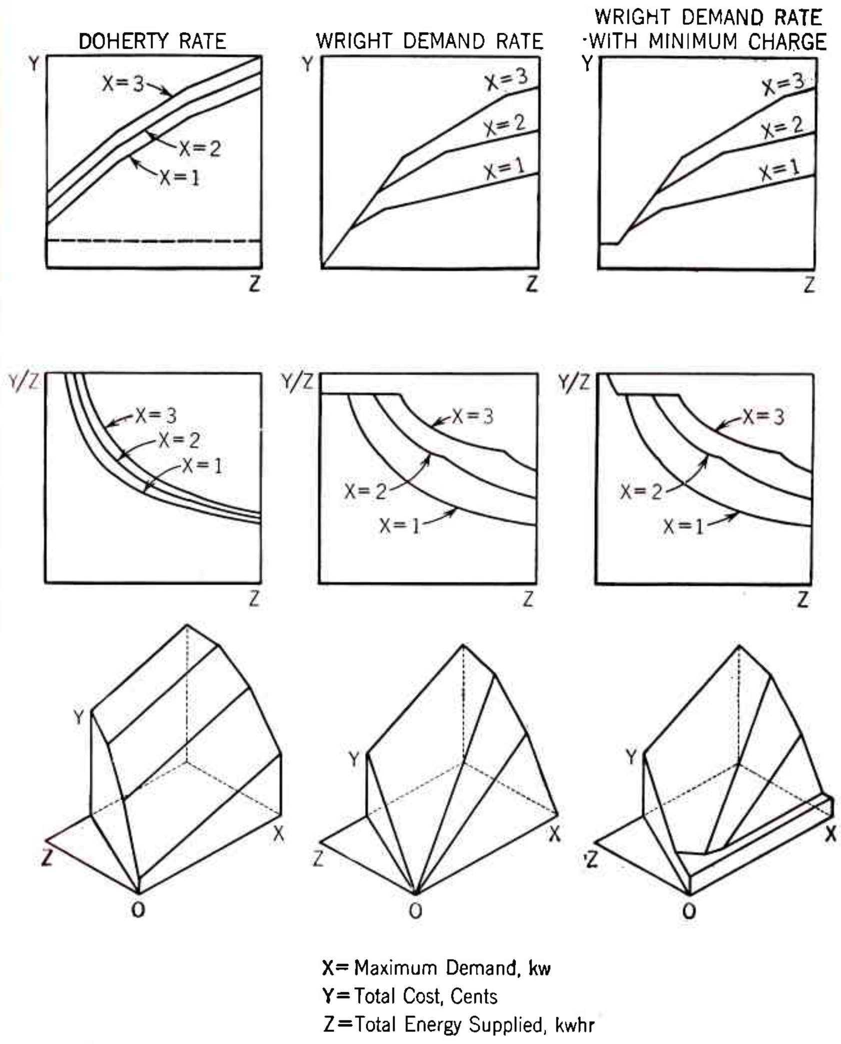
This method is usually most popular with the public since it implies a definite price for a concrete amount of a commodity.


MODIFIED STEP METER RATE

BLOCK METER RATE

HOPKINSON DEMAND RATE

FIG. 34-1 (*continued*). Basic forms of electrical-energy service rates.


DOHERTY RATE

WRIGHT DEMAND RATE

WRIGHT DEMAND RATE WITH MINIMUM CHARGE

FIG. 34-1 (*continued*). Basic forms of electrical-energy service rates.

The disadvantage of this rate lies in the fact that a consumer using no energy for a billing period has a zero charge, whereas he has incurred a definite expense to the company due to its readiness to serve him.

34-5. Step Meter Rate. The straight meter rate has the disadvantage of charging the same unit price for all magnitudes of energy consumption. It was demonstrated in Fig. 33-6 that increased generation (or consumption) spreads the item of fixed charges over a greater number of units of energy and, therefore, the price of the energy should decrease as the consumption increases. To rectify the inconsistency of the straight meter rate, several methods, sometimes termed "sliding scales," were introduced, of which the step meter rate was one of the two basic forms. The step meter rate is a group of straight meter rates of decreasing unit charges or slopes correspondingly applying to higher ranges of consumption. This rate can be expressed as follows:

$$
\begin{aligned}
y &= ez & &\text{where } 0 \leqq z \leqq a\\
y &= e_1 z_1 & &\text{where } a < z_1 \leqq b\\
y &= e_2 z_2 & &\text{where } b < z_2 \leqq c
\end{aligned}
$$

where $0, a, b, c$, etc. = limits of energy consumption
e, e_1, e_2, etc. = unit charges for various ranges of energy consumption

The equations are illustrated in Fig. 34-1, where all forms of rates are plotted for comparison.

An inconsistency develops in the total cost when the energy consumption nears the upper limit of one of the steps. By increasing the energy consumption so as just to enter the next step the total energy cost may be reduced. This can be corrected by a modification of the rate so that the total cost remains constant when changing from one unit price or step to the next lower unit price or step as shown in Fig. 34-1.

Example 34-1. A step meter rate is quoted as follows:

0–10 kwhr at 10 cents per kilowatthour
10–50 kwhr at 8 cents per kilowatthour
50–100 kwhr at 6 cents per kilowatthour
Over 100 kwhr at 5 cents per kilowatthour

To modify this rate so as to eliminate decreasing cost with increasing consumption, first find the total cost at the lower end of each step,

$$10 \times 8 = \$0.80 \qquad 50 \times 6 = \$3 \qquad 100 \times 5 = \$5$$

Then the upper end of each constant unit cost step is

$$^{80}\!/_{10} = 8 \text{ kwhr}$$
$$^{300}\!/_{8} = 37 \text{ kwhr}$$
$$^{500}\!/_{6} = 83 \text{ kwhr}$$

The modified step meter rate is then

0–8 kwhr at 10 cents per kilowatthour
8–10 kwhr for 80 cents total
10–37 kwhr at 8 cents per kilowatthour
37–50 kwhr for \$3 total
50–83 kwhr at 6 cents per kilowatthour
83–100 kwhr for \$5 total
Over 100 kwhr at 5 cents per kilowatthour

34-6. Block Meter Rate. This rate accomplishes the same purpose of decreasing unit energy charges with increasing consumption as the step meter rate without its aforementioned defect. In this rate a certain unit charge applies to a given *block* of energy, as contrasted to the *total* energy in the step meter rate. Then for succeeding blocks of energy the corresponding unit charges decrease.

The block meter rate can be reduced to the form

$$y = e_1z_1 + e_2z_2 + \cdots + e_n[z - (z_1 + z_2 + \cdots + z_{n-1})]$$

where $e_1, e_2, \ldots, e_n$ = unit energy charges for energy blocks of magnitudes

$$z_1 + z_2 + \cdots + z_n = z = \text{total energy consumption}$$

If the total energy consumption should fall within the second block, the equation would reduce to

$$y = e_1z_1 + e_2(z - z_1)$$

or

$$y = z_1(e_1 - e_2) + e_2z$$

Under this circumstance the first term is a constant, and the equation could be written

$$y = k + e_2z$$

Thus k is in the nature of a constant charge such as a customer charge if the total energy z always exceeds the first block z_1. For this circumstance the block meter rate can be looked upon as a modified form of the straight meter rate with an additional customer charge. The unit energy cost Y/Z (Fig. 34-1) is seen to approach the idealized unit cost variation of Fig. 33-6, which is the major objective of all rates.

Example 34-2. A block meter rate is quoted as follows:

First 10 kwhr at 10 cents per kilowatthour
Next 40 kwhr at 6 cents per kilowatthour
Next 50 kwhr at 4 cents per kilowatthour
Next 100 kwhr at 3 cents per kilowatthour
Excess over 200 kwhr at 2 cents per kilowatthour

Compute the total bills and average unit cost for consumption of 50, 100, 150, and 300 kwhr.

Solution

For 50 kwhr:
$y = (10 \times 0.1) + (40 \times 0.06)$ . . . . \$ 3.40
For 100 kwhr:
$y = (10 \times 0.1) + (40 \times 0.06) + (50 \times 0.04)$ . . . . \$ 5.40
For 150 kwhr:
$y = (10 \times 0.1) + (40 \times 0.06) + (50 \times 0.04) + (50 \times 0.03)$ . . . . \$ 6.90
For 300 kwhr:
$y = (10 \times 0.1) + (40 \times 0.06) + (50 \times 0.04) + (100 \times 0.03) + (100 \times 0.02)$ \$10.40

Total kwhr	Total cost	Unit cost, cents per kwhr
50	\$ 3.40	6.8
100	5.40	5.4
150	6.90	4.6
300	10.40	2.9

34-7. Hopkinson Demand Rate. In either a straight meter rate or a flat demand rate the actual costs cannot be adequately allocated; this is due to trying to charge on the basis of a single item when actually the costs vary with three items. Partly recognizing this fact, Dr. John Hopkinson in 1892 proposed a rate system explicitly including demand and energy charges. This rate form is expressed as

$$y = dx + ez$$

Both the demand and energy charges may be graduated on either a block or a step system. This rate introduces the problem of measuring a consumer's demand. In some instances the connected load multiplied by a representative demand factor determines the billing demand. In others, meters continuously record the cumulative energy consumption for stated intervals such as 5 min, 15 min, 30 min, or 1 hr. The maximum cumulative consumption is then the maximum demand for the billing period. The shorter the period over which the cumulative energy demand is measured, the higher the maximum demand owing to the fluctuating nature of most loads. Some utility companies specify a billing demand in terms of the number of rooms in a building for residential service; others arbitrarily determine billing demand upon the basis of floor area. Wherever billing demands are computed by some arbitrary standard, they must be paid for even if no energy is consumed during the period. This requirement theoretically disappears if the demand is metered. However, it is usual to specify a minimum demand that must be paid for.

Example 34-3. A Hopkinson demand rate is quoted as follows:

Demand:

First kilowatt of maximum demand at \$6 per kilowatt per month
Next 4 kw of maximum demand at \$5 per kilowatt per month
Excess over 5 kw of maximum demand at \$4 per kilowatt per month

Energy:

First 50 kwhr at 6 cents per kilowatthour
Next 50 kwhr at 4 cents per kilowatthour
Next 200 kwhr at 3 cents per kilowatthour
Next 400 kwhr at 2.5 cents per kilowatthour
Excess over 700 kwhr at 2 cents per kilowatthour

(*a*) Compute the monthly bill for a total consumption of 1,500 kwhr and a maximum demand of 12 kw. Find the unit energy cost.

(*b*) If the month contains 30 days, what is the lowest possible bill for the given energy consumption? Compute the corresponding unit energy cost.

Solution

(*a*) Demand charge $= 6 + (4 \times 5) + (7 \times 4)$ \$54
Energy charge $= (50 \times 0.06) + (50 \times 0.04) + (200 \times 0.03) + (400 \times 0.05) + (800 \times 0.02)$ 37
Total monthly bill \$91

$$\text{Unit energy cost, } \frac{9,100}{1,500} = 6.07 \text{ cents per kwhr}$$

(*b*) The lowest possible bill occurs when the demand is held to a minimum which occurs at 100 per cent load factor. For this condition

$$\text{Average load} = \text{maximum load} = \frac{1,500}{30 \times 24} = 2.083 \text{ kw}$$

Then

Demand charge $= 6 + (1.083 \times 5)$ \$11.42
Energy charge 37.00
Lowest possible bill \$48.42

$$\text{Unit energy cost} = \frac{4,842}{1,500} = 3.22 \text{ cents per kwhr}$$

Since in most systems the peak occurs once a year, it is frequently specified that the billing demand shall be not less than a specified minimum value, but if the metered demand in any month exceeds this minimum, the highest value metered shall be the billing demand for the succeeding 12 months. At the expiration of the 12 months the minimum value again controls unless the metered demand is higher than the minimum, when such higher value is again used as the billing demand for another 12 months or until it is displaced by a succeeding higher value.

34-8. Doherty Rate. This rate, also called the "three-part charge" rate, was introduced by Henry L. Doherty at the beginning of the twentieth century. It is an extension of the Hopkinson rate in that it adds the customer charge, the form being identical with the basic form mentioned previously.

$$y = dx + ez + c$$

In this form, for zero energy consumption but connection to the source of energy still in effect, the consumer will pay the customer charge. The Doherty rate is sometimes modified by specifying the minimum demand and the minimum energy consumption that must be paid for if they are less than the minimum values specified. In this manner the customer charge c is incorporated with the demand and energy component.

34-9. Wright Demand Rate. This rate was first proposed in 1896 by Arthur Wright at Brighton, England. While the Hopkinson demand rate offers explicit inducement to a consumer to keep his demand at a minimum through lowering the demand charge, the Wright demand rate intensifies this inducement by lowering *both* the demand and the energy charge for a reduction in maximum demand or, in other words, an improvement in load factor.

This rate may be expressed in the form

$$y = e_1t_1d + e_2t_2d + e_3t_3d + \cdots + e_n[z - (t_1 + t_2 + t_3 + \cdots + t_{n-1})d]$$

where $t_1, t_2, t_3, \ldots, t_n$ = specified succeeding blocks of time, hr, total of all blocks being equal to number of hours in billing period

d = maximum demand during billing period

z = total energy consumption during billing period

$e_1, e_2, e_3, \ldots, e_n$ in decreasing order of magnitude = energy charges for corresponding blocks of energy $t_1d, t_2d, t_3d, \ldots, t_nd$

This rate is usually specified for industrial consumers, who have some measure of control over their maximum demands. The rate is often modified by stating a minimum charge which must be paid if the energy for the billing period falls below the amount allowed by such charge.

Example 34-4. A Wright demand rate is quoted as follows:

For energy purchased monthly, the equivalent of

1. The first 50 hr use of maximum demand at 6 cents per kilowatthour
2. The next 50 hr use of maximum demand at 4 cents per kilowatthour
3. The next 100 hr use of maximum demand at 3 cents per kilowatthour
4. The next 200 hr use of maximum demand at 2 cents per kilowatthour
5. All energy in excess of foregoing blocks at 1 cent per kilowatthour

(*a*) Compute the bill for a monthly energy consumption of 432,000 kwhr with a maximum demand of 1,600 kw. Compute the unit energy cost corresponding.

(*b*) Find the lowest possible bill for this energy consumption and the corresponding unit energy cost for a 30-day month.

Solution

(*a*)	1st block:	50 × 1,600 × 0.06 = 80,000 × 0.06	$ 4,800
	2d block:	50 × 1,600 × 0.04 = 80,000 × 0.04	3,200
	3d block:	100 × 1,600 × 0.03 = 160,000 × 0.03	4,800
	4th block:	112,000 × 0.02	2,240
	Total	432,000 kwhr	$15,040

$$\text{Unit energy cost} = \frac{1{,}504{,}000}{432{,}000} = 3.48 \text{ cents per kwhr}$$

(*b*) The lowest possible bill occurs at 100 per cent load factor. Then

$$\text{Average load} = \text{maximum load} = \frac{432{,}000}{30 \times 24} = 600 \text{ kw}$$

Then

1st block:	50 × 600 × 0.06 = 30,000 × 0.06	$1,800
2d block:	50 × 600 × 0.04 = 30,000 × 0.04	1,200
3d block:	100 × 600 × 0.03 = 60,000 × 0.03	1,800
4th block:	200 × 600 × 0.02 = 120,000 × 0.02	2,400
5th block:	192,000 × 0.01	1,920
Total	432,000 kwhr	$9,120

$$\text{Unit energy cost} = \frac{912{,}000}{432{,}000} = 2.11 \text{ cents per kwhr}$$

34-10. Rate Adjustments. The constants used in rate forms are based on certain specified conditions of operation and prices. Owing to the monopolistic character of the electric utilities, they are allowed a "fair return" on their investment. If conditions of operations and prices should so act as to reduce the operating costs of the utility, it is required to pass along such reductions to the consumer and limit itself to its stated fair return; conversely, if operating costs should increase, it will be allowed to pass along such increases within the limitations of the allowed fair return. Such adjustments must be set up in the rate forms. Some of the adjustments found in practice are:

1. *Fuel-price adjustment.* This provides for increasing or lowering the total bill as actual fuel price deviates from standard price. This sometimes includes the change in efficiency of generation from a given standard of efficiency.
2. *Hydraulic-fuel adjustment.* This provides for variation in fuel consumption as affected by availability of water power.
3. *Tax adjustment.* This provides for variation in tax costs of a utility.
4. *Wage adjustment.* This provides for variation in labor costs of a utility.
5. *Power-factor adjustment.* This provides a penalty for a consumer if the power factor of load drops below a specified standard.
6. *Discount adjustment.* This provides a discount for a consumer for the prompt payment of bills.

34-11. Rate Design. One of the most controversial subjects is that of cost allocation to the various consumers served by a central-station service. Although there is general agreement as to the justice of the general rate form as suggested by the Doherty rate, the details for determining the magnitudes of the constants for the various types of consumers depend upon what is considered an equitable distribution of costs. Without holding any brief for a particular method of allocation, some of the methods of allocation are described in the following to illustrate the nature of the problem.

The principal advantage of the scheme of central-station service lies in the reduction of investment in generating facilities made possible by the diversity in maximum demands of the individual consumers relative to each other. Equitable allocation of the fixed charges on this investment among the consumers then arises.

One method known as the *peak-responsibility* allocation assumes that the cost should be distributed according to the proportion in which each consumer requires energy at the time of the system peak demand. The main disadvantages of this method lie in the facts that peaks of a central-station system do not occur always at the same time, that proportions of consumer demands for successive peaks vary, and that any consumers that do not require energy at the system peak time but do require firm power at other times of the day are not sharing an equitable portion of the fixed charges.

TABLE 34-1. CONSUMER AND SYSTEM LOADS

Time	System total, mw	Consumer load, mw			
		A	*B*	*C*	*D*
1 A.M.	40	10	20	10	0
2	40	10	10	10	10
3	50	10	10	10	20
4	50	10	10	10	20
5	40	10	10	10	10
6	40	10	20	10	0
7	60	10	40	10	0
8	70	10	50	10	0
9	70	10	50	10	0
10	70	10	50	10	0
11	70	10	50	10	0
12 M.	70	10	50	10	0
1 P.M.	50	10	30	10	0
2	70	10	50	10	0
3	70	10	50	10	0
4	90	10	50	30	0
5	100	10	30	60	0
6	90	10	20	60	0
7	80	10	10	60	0
8	70	10	10	50	0
9	60	10	10	40	0
10	50	10	10	30	0
11	40	10	10	20	0
12	30	10	10	10	0
Total	1,470	240	660	510	60

Another method known as the *noncoincident-demand* allocation attempts to remedy these defects by proportioning the fixed charges according to the consumer individual maximum demands regardless of the time when they occur. The difference in allocating costs under the two methods is illustrated for an extremely simplified case. Table 34-1 shows the daily load variation and the individual demands of the consumers. Assume that this curve repeats itself daily. Table 34-2 shows the percentage of the fixed charges that would be allocated to each consumer under each method of apportionment.

TABLE 34-2. COMPARISON OF ALLOCATIONS OF FIXED CHARGES—PEAK RESPONSIBILITY VS. NONCOINCIDENT DEMAND

Consumer	Peak responsibility		Noncoincident demand	
	Mw	Per cent of fixed charges	Mw	Per cent of fixed charges
A	10	10	10	7.1
B	30	30	50	35.8
C	60	60	60	42.9
D	0	0	20	14.2
Total.	100	100	140	100.0

Under peak-responsibility allocation consumer *D* escapes bearing a fair share of investment charges. At the other extreme consumer *C* bears the maximum proportion since his peak occurs at the time of the system peak. Under noncoincident demand consumer *D* bears a part of the fixed charges in proportion to his maximum requirements, due consideration being given the relative diversity of the demands. Consumers *A* and *C* gain in economy, while *B* and *D* assume a fairer share of the investment costs under noncoincident demand as compared with peak responsibility.

However, consumer *A* has a 100 per cent load factor and therefore cannot have a diversity in respect to any other load. Under noncoincident-demand allocation *A* benefits from a diversity that does not exist; i.e., if *A* were removed from the system, it would be possible to eliminate 10 mw of capacity without disturbing the requirements of the remaining consumers. Hence *A* should pay the full investment charges on its identical average and maximum demand and not enjoy a discount at the expense of the other consumers. To make diversity possible, the load factor of the consumers must be less than 100 per cent. Furthermore, the lower the load factor, the greater the benefits that become possible because of diversity.

A method designated as the "two-element" method endeavors to account for the relationship of load factor and diversity credit. It is illustrated by a numerical example.

Example 34-5. The central station supplying the system in Table 34-1 has a capacity of 120 mw costing $100 per kilowatt upon which annual fixed charges of $1,440,000 must be distributed. Since the maximum system load is 100 mw, the 20-mw difference from station capacity represents reserve capacity. For this example it will be assumed that the fixed charges on the reserve capacity amounting to $240,000 are allocated on the basis of a customer charge. Then the active capacity equal to the system peak load of 100 mw must earn fixed charges annually amounting to $1,200,000.

If the 100-mw portion of the station capacity operated at an annual load factor of 100 per cent, its fixed charges could be recovered entirely by making an energy charge of 1,200,000/(8,760 × 100,000) = $0.00137, or 0.137 cent per kilowatthour.

With the actual daily load of Table 34-1 obtaining for 365 days a year the cost recovered by this energy rate would be only 1,470,00 × 365 × 0.00137 = $735,100, leaving an unrecovered cost of $464,900. Now the average load on the system for a year (or day) is 1,470,000/24 = 61,250 kw. On an equivalent basis it may be assumed that the first 61,250 kw of capacity earns its fixed charges by being loaded 100 per cent of the time and being charged for on the basis of the energy output. The remaining capacity of 100,000 − 61,250 = 38,750 kw is necessary for variation of demand above the average. The cost of this capacity, which is $464,900, must then be equally spread over the units of maximum demand of each consumer that is in excess of the consumer's average demand. From Table 34-2 the total noncoincident demands of all consumers is 140,000 kw, which compares with the average demand of 61,250 kw. The excess demand of 140,000 − 61,250 = 78,750 kw must then recover the remaining $464,900 of fixed charges by a demand charge of $464,900/78,750 = $5.90 per kilowatt. This rate could then be expressed in the form of 0.137 cent per kilowatthour maximum demand in excess of average load.

From a practical standpoint it is more convenient to base the demand charge on the total maximum demand rather than on the excess figure. Recognizing that (0.137 cent per kilowatthour) × (8,760 kwhr per kw) = $12 per kilowatt, it is immaterial whether the energy charge *or* the demand charge is used to recover the same portion of the fixed charges. To use the same demand figure for the total maximum demand kilowatts as for the excess kilowatts above average demand, it follows that the equivalent of either 0.137 cent per kilowatthour *or* $12 per kilowatt is $5.90 per kilowatt plus (1,200−590)/8,760 = 0.0697 cent per kilowatthour. That is, the fixed charges on the active capacity will all be recovered by billing on the annual basis of $5.90 per kilowatt of maximum demand plus 0.0697 cent per kilowatthour.

The allocation of fixed charges among the consumers of Table 34-1 on the above rate is shown in Table 34-3 and is compared with the other methods of allocation.

By comparing the allocation of fixed charges among the consumers for the two-element method of charging and the noncoincident-demand method the 100 per cent load factor of *A* is fully accounted for in the former in comparison with the unjustified credit assigned it for diversity in the latter method. The 55 per cent load factor of *B* causes it to be assigned a somewhat larger portion of the fixed charges in the two-element method, whereas the 35.4 per cent load factor of *C* and the 12.5 per cent load factor of *D* cause a smaller portion of fixed charges to be assigned to them.

Assuming that the two-element method is used for the rate design, it is then necessary to allocate the fixed charges on the reserve capacity. If this is done on a customer basis, there will be an annual charge of $240,000/4 = $60,000 per customer per year. If it is considered more equitable to distribute on the noncoincident-demand basis, there will be an additional demand charge of $240,000/140,000 kw = $1.71 per kilowatt maximum demand.

TABLE 34-3. COMPARISON OF DEMAND-CHARGE ALLOCATIONS

Consumer	Two-element method of recovering fixed charges				Per cent of fixed charges		
	Maximum demand, mw	Annual maximum demand charge			Two-element method	Noncoincident demand	Peak responsibility
		$5.90 per kw	0.0697 cents per kwhr	Total			
A	10	$ 59,000	$ 61,100	$ 120,100	10.0	7.1	10.0
B	50	295,000	168,000	463,000	38.6	35.8	30.0
C	60	354,000	129,700	483,700	40.3	42.9	60.0
D	20	118,000	15,200	133,200	11.1	14.2	0.0
Total........	..	$826,000	$374,000	$1,200,000	100.0	100.0	100.0

For the latter condition the total rate for recovering fixed charges on station investment would be $7.61 per kilowatt of maximum demand plus 0.0697 cent per kilowatthour. If the total production cost of the station is $2,000,000, a detailed analysis of each item of operating cost might show that $700,000 was a function of demand and the remaining $1,300,000 a function of energy produced. The demand rate for recovery of operating cost is then $700,000/140,000 = $5 per kilowatt of maximum demand, and the energy rate is $1,300,000/(365 × 1,470,000) = $0.00242, or 0.242 cent per kilowatthour. For this particular example, then, the total rate at the station bus is

1. Fixed charges of active capacity......	$ 5.90 per kw	plus 0.0697 cent per kwhr
2. Fixed charges of reserve capacity.....	1.71 per kw	
3. Operating costs.....................	5.00 per kw	plus 0.242 cent per kwhr
4. Total rate at station bus.............	$12.61 per kw	plus 0.3117 cent per kwhr

This elementary example only illustrates some of the problems of rate design, and the student is cautioned against assuming it as typical. Rate designing for a public utility is much more complex than indicated, owing to the variety of equipment and services that such a property must supply. In round figures the proportion of relative investment in capital equipment of some companies might be found to approximate the data of the following tabulation:

	Per cent
1. Central generating stations..........	35
2. Transmission equipment.............	15
3. Substations........................	15
4. Distribution system................	25
5. Offices, garages, shops, etc..........	10
6. Total............................	100

The layman often wonders about the wide margin in the cost of producing a kilowatt at the station bus (about 0.5 cent) and the price he pays for use in his home (about 5 cents). This comes about from the enormous amount of equipment required for operating and managing the business as a whole and the personnel required for the commercial aspects. A breakdown of the various items of cost would be about as appears in the following tabulation for some companies:

		Per cent	
1.	Generating-station investment charges	11	
2.	Transmission-equipment investment charges	5	
3.	Substation investment charges	6	
4.	Distribution-equipment investment charges	28	
5.	Miscellaneous investment charges	2	
6.	Total investment charges		52
7.	Operating costs of power plants, transmission system, and substations	12	
8.	Distribution-system operation	5	
9.	Meter reading, billing, collection, sales, etc	31	
10.	Total operating costs		48
11.	Total cost of service		100

PROBLEMS

34-1. An industrial concern purchases energy under the following rate:

For energy purchased monthly the equivalent of

1. The first 30 hr use of the maximum demand at 5 cents per kilowatthour
2. The next 100 hr use of the maximum demand at 2 cents per kilowatthour
3. The next 200 hr use of the maximum demand at 1 cent per kilowatthour
4. All energy in excess of foregoing blocks at 0.8 cent per kilowatthour

The total monthly charge shall be not less than $3,000.

a. Calculate the monthly bill for energy consumed of 420,000 kwhr with a maximum demand of 2,000 kw.

b. Calculate the bill for a maximum demand of 1,000 kw with the same total energy.

34-2. A step meter rate is quoted in Table P-1.

Table P-1

Monthly consumption limits, kwhr	*Rate per kwhr, cents*
0–100	10
101–200	9
201–300	8
301 and up	7

a. Calculate the monthly bill for 200 kwhr.

b. Calculate the monthly bill for 201 kwhr.

c. Find the range of kilowatthours at the beginning of each step at which the monthly bill would be less than the bill for the maximum consumption in the preceding step.

34-3. A common form of residential rate appears as follows:

First 10 kwhr per month or less at $1
Next 40 kwhr per month at 7 cents per kilowatthour
Next 50 kwhr per month at 5 cents per kilowatthour
Next 100 kwhr per month at 3 cents per kilowatthour
All kilowatthours in excess of 200 kwhr at 2 cents per kilowatthour

a. Calculate the monthly bill for 8 kwhr consumption.
b. Calculate the monthly bill for 85 kwhr consumption.
c. Calculate the monthly bill for 300 kwhr consumption.
d. Find the average unit cost of 1 kwhr for (*a*), (*b*), and (*c*).
e. Identify the form of the rate.

34-4. A form often used for commercial consumers of energy is stated as follows:

1. A charge of $4 per month per kilowatthour of maximum demand
2. Plus 6 cents per kilowatthour for the first 100 kwhr per month
   5 cents per kilowatthour for the next 200 kwhr per month
   3 cents per kilowatthour for the next 300 kwhr per month
   2 cents per kilowatthour for energy in excess of 600 kwhr
3. Plus $2 per month

Calculate the bill for a monthly consumption of 800 kwhr with a maximum demand of 5 kw.

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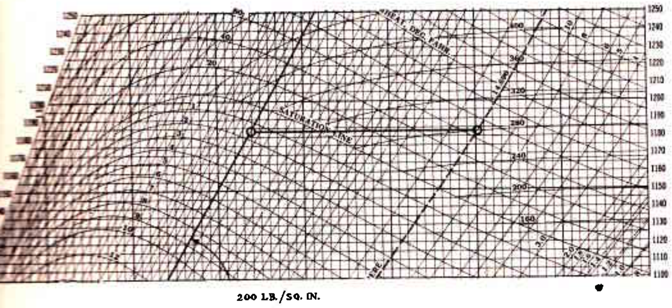
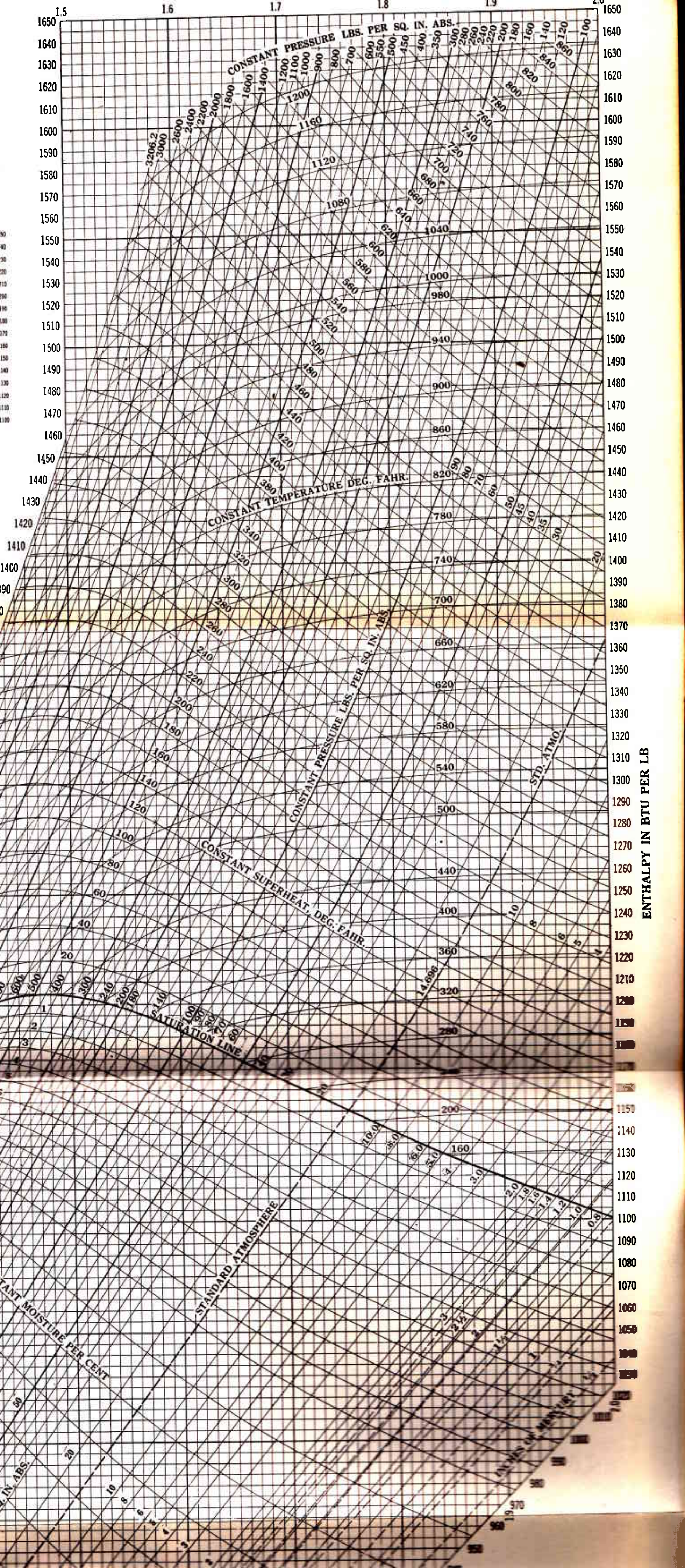
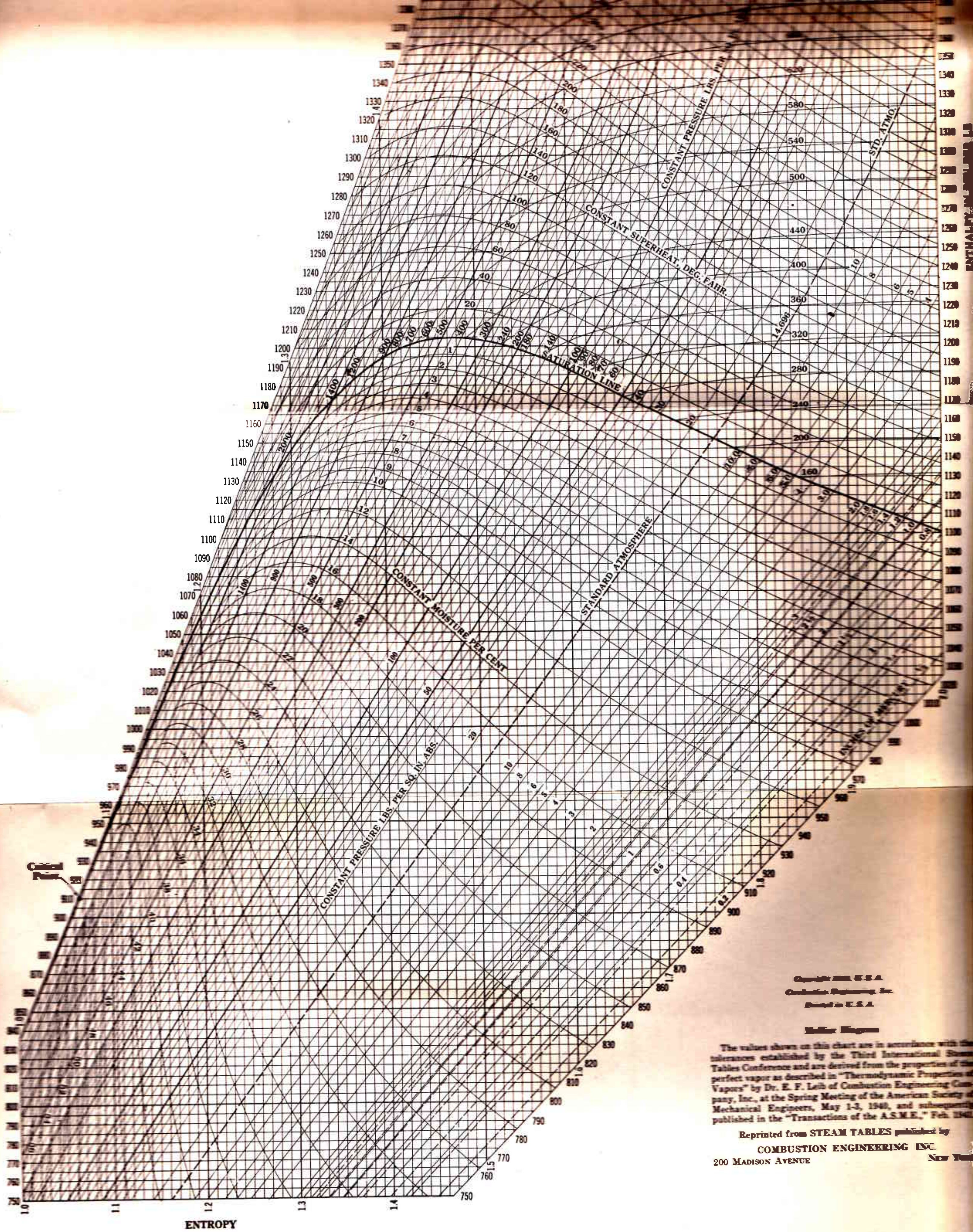


How to Use the Mollier Diagram to Determine Quality from Calorimeter Data.

Pressure in calorimeter, atmospheric (14.696 lb. per sq. in. abs.).
Temperature in calorimeter, 280 deg. F.
Pressure in steam main, 200 lb. per sq. in. abs.
Per cent moisture from chart, 1.7 per cent.
Quality, 98.3 per cent.





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Mollier Diagram

The values shown on this chart are in accordance with th tolerances established by the Third International Stea Tables Conference and are derived from the properties of th perfect vapor as described in "Thermodynamic Properties o Vapors" by Dr. E. F. Leib of Combustion Engineering Com pany, Inc., at the Spring Meeting of the American Society o Mechanical Engineers, May 1-3, 1940, and subsequently published in the "Transactions of the A.S.M.E.," Feb. 194

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Fig. 1. Mollier diagram of thermodynamic properties of steam.

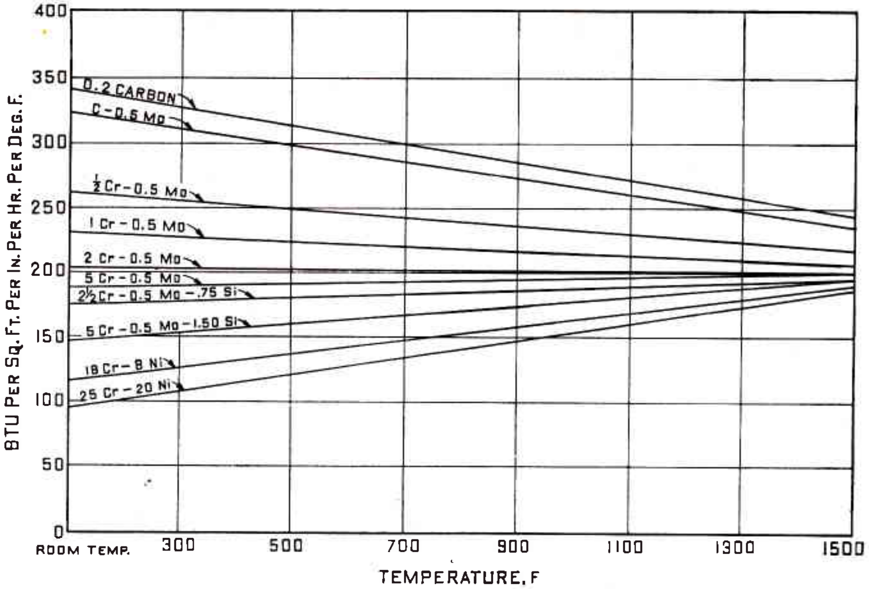


FIG. 2. Thermal conductivities of various steels (k = Btu per hr-sq ft-°F-in.). (*Courtesy of National Tube Division, United States Steel Corporation, Bulletin* 26.)

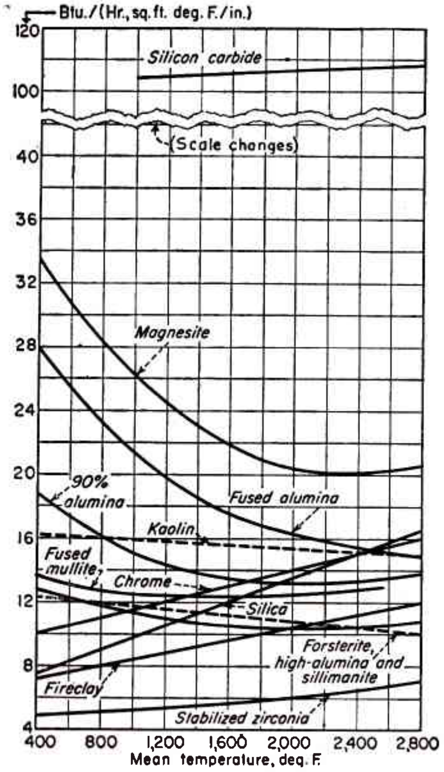


FIG. 3. Thermal conductivities of refractory brick (k = Btu per hr-sq ft-°F-in.). (*Courtesy of C. L. Norton, Refractories for Every Use, Chemical Engineering, June,* 1952.)

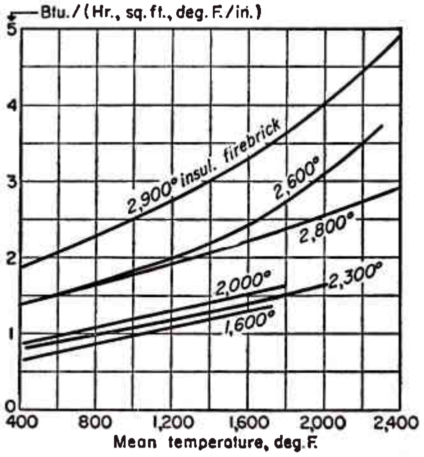
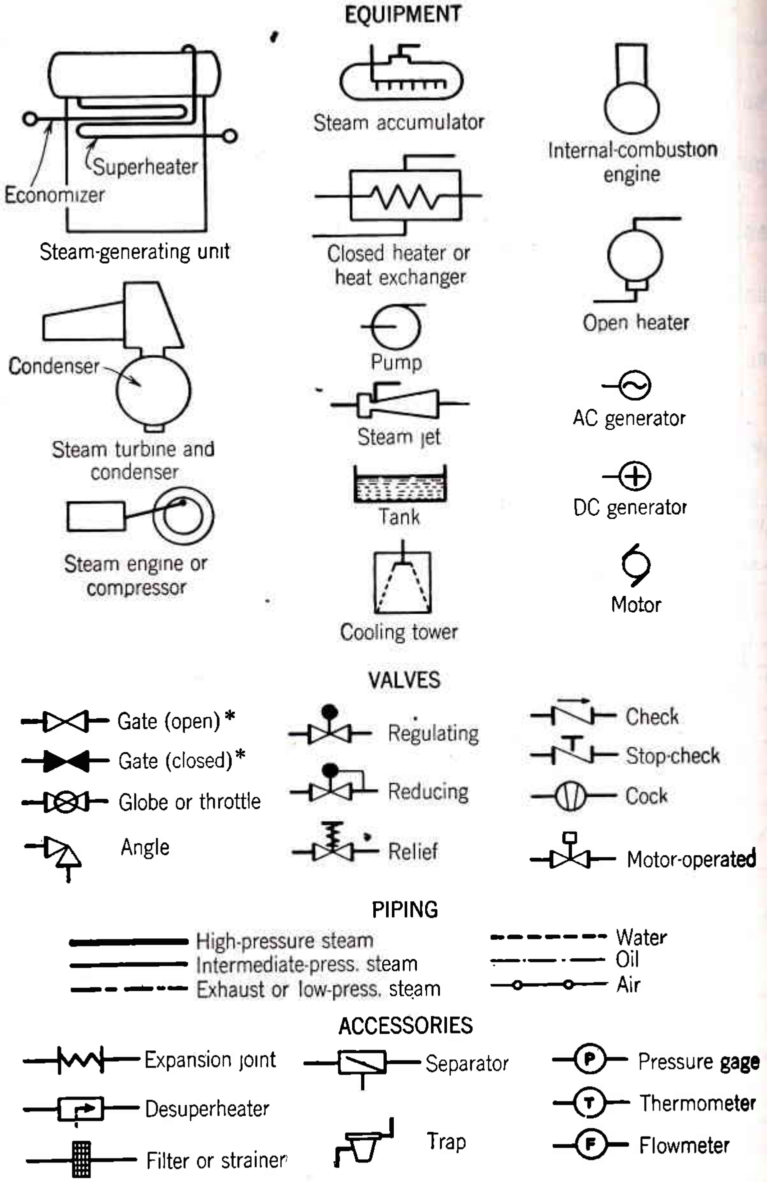


FIG. 4. Thermal conductivities of insulating firebrick (k = Btu per hr-sq ft-°F-in.). (*Courtesy of C. L. Norton, Refractories for Every Use, Chemical Engineering, June,* 1952.)



* Filled-in triangles show any valve normally closed, outline triangles show it open.

FIG. 5. Symbols suggested for use in power-plant heat-balance drawings.

TABLE 1. THERMODYNAMIC PROPERTIES OF STEAM

Dry saturated steam: pressure table[a]

Abs. press., psia	Temp., F	Specific volume		Enthalpy			Entropy			Internal energy		Abs. press., psia
		Sat. liquid	Sat. vapor	Sat. liquid	Evap.	Sat. vapor	Sat. liquid	Evap.	Sat. vapor	Sat. liquid	Sat. vapor	
p	t	v_f	v_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g	u_f	u_g	p
1.0	101.74	0.01614	333.6	69.70	1036.3	1106.0	0.1326	1.8456	1.9782	69.70	1044.3	**1.0**
2.0	126.08	0.01623	173.73	93.99	1022.2	1116.2	0.1749	1.7451	1.9200	93.98	1051.9	**2.0**
3.0	141.48	0.01630	118.71	109.37	1013.2	1122.6	0.2008	1.6855	1.8863	109.36	1056.7	**3.0**
4.0	152.97	0.01636	90.63	120.86	1006.4	1127.3	0.2198	1.6427	1.8625	120.85	1060.2	**4.0**
5.0	162.24	0.01640	73.52	130.13	1001.0	1131.1	0.2347	1.6094	1.8441	130.12	1063.1	**5.0**
6.0	170.06	0.01645	61.98	137.96	996.2	1134.2	0.2472	1.5820	1.8292	137.94	1065.4	**6.0**
7.0	176.85	0.01649	53.64	144.76	992.1	1136.9	0.2581	1.5586	1.8167	144.74	1067.4	**7.0**
8.0	182.86	0.01653	47.34	150.79	988.5	1139.3	0.2674	1.5383	1.8057	150.77	1069.2	**8.0**
9.0	188.28	0.01656	42.40	156.22	985.2	1141.4	0.2759	1.5203	1.7962	156.19	1070.8	**9.0**
10	193.21	0.01659	38.42	161.17	982.1	1143.3	0.2835	1.5041	1.7876	161.14	1072.2	**10**
14.696	212.00	0.01672	26.80	180.07	970.3	1150.4	0.3120	1.4446	1.7566	180.02	1077.5	**14.696**
15	213.03	0.01672	26.29	181.11	969.7	1150.8	0.3135	1.4415	1.7549	181.06	1077.8	**15**
20	227.96	0.01683	20.089	196.16	960.1	1156.3	0.3356	1.3962	1.7319	196.10	1081.9	**20**
25	240.07	0.01692	16.303	208.42	952.1	1160.6	0.3533	1.3606	1.7139	208.34	1085.1	**25**
30	250.33	0.01701	13.746	218.82	945.3	1164.1	0.3680	1.3313	1.6993	218.73	1087.8	**30**
35	259.28	0.01708	11.898	227.91	939.2	1167.1	0.3807	1.3063	1.6870	227.80	1090.1	**35**
40	267.25	0.01715	10.498	236.03	933.7	1169.7	0.3919	1.2844	1.6763	235.90	1092.0	**40**
45	274.44	0.01721	9.401	243.36	928.6	1172.0	0.4019	1.2650	1.6669	243.22	1093.7	**45**
50	281.01	0.01727	8.515	250.09	924.0	1174.1	0.4110	1.2474	1.6585	249.93	1095.3	**50**
55	287.07	0.01732	7.787	256.30	919.6	1175.9	0.4193	1.2316	1.6509	256.12	1096.7	**55**
60	292.71	0.01738	7.175	262.09	915.5	1177.6	0.4270	1.2168	1.6438	261.90	1097.9	**60**
65	297.97	0.01743	6.655	267.50	911.6	1179.1	0.4342	1.2032	1.6374	267.29	1099.1	**65**
70	302.92	0.01748	6.206	272.61	907.9	1180.6	0.4409	1.1906	1.6315	272.38	1100.2	**70**
75	307.60	0.01753	5.816	277.43	904.5	1181.9	0.4472	1.1787	1.6259	277.19	1101.2	**75**
80	312.03	0.01757	5.472	282.02	901.1	1183.1	0.4531	1.1676	1.6207	281.76	1102.1	**80**
85	316.25	0.01761	5.168	286.39	897.8	1184.2	0.4587	1.1571	1.6158	286.11	1102.9	**85**
90	320.27	0.01766	4.896	290.56	894.7	1185.3	0.4641	1.1471	1.6112	290.27	1103.7	**90**
95	324.12	0.01770	4.652	294.56	891.7	1186.2	0.4692	1.1376	1.6068	294.25	1104.5	**95**
100	327.81	0.01774	4.432	298.40	888.8	1187.2	0.4740	1.1286	1.6026	298.08	1105.2	**100**
110	334.77	0.01782	4.049	305.66	883.2	1188.9	0.4832	1.1117	1.5948	305.30	1106.5	**110**
120	341.25	0.01789	3.728	312.44	877.9	1190.4	0.4916	1.0962	1.5878	312.05	1107.6	**120**
130	347.32	0.01796	3.455	318.81	879.9	1191.7	0.4995	1.0817	1.5812	318.38	1108.6	**130**
140	353.02	0.01802	3.220	324.82	868.2	1193.0	0.5069	1.0682	1.5751	324.35	1109.6	**140**
150	358.42	0.01809	3.015	330.51	863.6	1194.1	0.5138	1.0556	1.5694	330.01	1110.5	**150**
160	363.53	0.01815	2.834	335.93	859.2	1195.1	0.5204	1.0436	1.5640	335.39	1111.2	**160**
170	368.41	0.01822	2.675	341.09	854.9	1196.0	0.5266	1.0324	1.5590	340.52	1111.9	**170**
180	373.06	0.01827	2.532	346.03	850.8	1196.9	0.5325	1.0217	1.5542	345.42	1112.5	**180**
190	377.51	0.01833	2.404	350.79	846.8	1197.6	0.5381	1.0116	1.5497	350.15	1113.1	**190**
200	381.79	0.01839	2.288	355.36	843.0	1198.4	0.5435	1.0018	1.5453	354.68	1113.7	**200**
250	400.95	0.01865	1.8438	376.00	825.1	1201.1	0.5675	0.9588	1.5263	375.14	1115.8	**250**
300	417.33	0.01890	1.5433	393.84	809.0	1202.8	0.5879	0.9225	1.5104	392.79	1117.1	**300**
350	431.72	0.01913	1.3260	409.69	794.2	1203.9	0.6056	0.8910	1.4966	408.45	1118.0	**350**
400	444.59	0.0193	1.1613	424.0	780.5	1204.5	0.6214	0.8630	1.4844	422.6	1118.5	**400**
450	456.28	0.0195	1.0320	437.2	767.4	1204.6	0.6356	0.8378	1.4734	435.5	1118.7	**450**
500	467.01	0.0197	0.9278	449.4	755.0	1204.4	0.6487	0.8147	1.4634	447.6	1118.6	**500**
550	476.94	0.0199	0.8424	460.8	743.1	1203.9	0.6608	0.7934	1.4542	458.8	1118.2	**550**
600	486.21	0.0201	0.7698	471.6	731.6	1203.2	0.6720	0.7734	1.4454	469.4	1117.7	**600**
650	494.90	0.0203	0.7083	481.8	720.5	1202.3	0.6826	0.7548	1.4374	479.4	1117.1	**650**
700	503.10	0.0205	0.6554	491.5	709.7	1201.2	0.6925	0.7371	1.4296	488.8	1116.3	**700**
750	510.86	0.0207	0.6092	500.8	699.2	1200.0	0.7019	0.7204	1.4223	598.0	1115.4	**750**
800	518.23	0.0209	0.5687	509.7	688.9	1198.6	0.7108	0.7045	1.4153	506.6	1114.4	**800**
850	525.26	0.0210	0.5327	518.3	678.8	1197.1	0.7194	0.6891	1.4085	515.0	1113.3	**850**
900	531.98	0.0212	0.5006	526.6	668.8	1195.4	0.7275	0.6744	1.4020	523.1	1112.1	**900**
950	538.43	0.0214	0.4717	534.6	659.1	1193.7	0.7355	0.6602	1.3957	530.9	1110.8	**950**
1000	544.61	0.0216	0.4456	542.4	649.4	1191.8	0.7430	0.6467	1.3897	538.4	1109.4	**1000**
1100	556.31	0.0220	0.4001	557.4	630.4	1187.8	0.7575	0.6205	1.3780	552.9	1106.4	**1100**
1200	567.22	0.0223	0.3619	571.7	611.7	1183.4	0.7711	0.5956	1.3667	566.7	1103.0	**1200**
1300	577.46	0.0227	0.3293	585.4	593.2	1178.6	0.7840	0.5719	1.3559	580.0	1099.4	**1300**
1400	587.10	0.0231	0.3012	598.7	574.7	1173.4	0.7963	0.5491	1.3454	592.7	1095.4	**1400**
1500	596.23	0.0235	0.2765	611.6	556.3	1167.9	0.8082	0.5269	1.3351	605.1	1091.2	**1500**
2000	635.82	0.0257	0.1878	671.7	463.4	1135.1	0.8619	0.4230	1.2849	662.2	1065.6	**2000**
2500	668.13	0.0287	0.1307	730.6	360.5	1091.1	0.9126	0.3197	1.2322	717.3	1030.6	**2500**
3000	695.36	0.0346	0.0358	802.5	217.8	1020.3	0.9731	0.1885	1.1615	783.4	972.7	**3000**
3206.2	705.40	0.0503	0.0503	902.7	0	902.7	1.0580	0	1.0580	872.9	872.9	**3206.2**

[a] Abridged from "Thermodynamic Properties of Steam" by Joseph H. Keenan and Frederick G. Keyes, John Wiley & Sons, Inc., New York, 1937.

TABLE 2. THERMODYNAMIC PROPERTIES OF STEAM

Dry saturated steam: temperature table[a]

Temp., F	Abs. press., psia	Specific volume			Enthalpy			Entropy			Temp., F
t	p	Sat. liquid v_f	Evap. v_{fg}	Sat. vapor v_g	Sat. liquid h_f	Evap. h_{fg}	Sat. vapor h_g	Sat. liquid s_f	Evap. s_{fg}	Sat. vapor s_g	t
32	0.08854	0.01602	3306	3306	0.00	1075.8	1075.8	0.0000	2.1877	2.1877	**32**
35	0.09995	0.01602	2947	2947	3.02	1074.1	1077.1	0.0061	2.1709	2.1770	**35**
40	0.12170	0.01602	2444	2444	8.05	1071.3	1079.3	0.0162	2.1435	2.1597	**40**
45	0.14752	0.01602	2036.4	2036.4	13.06	1068.4	1081.5	0.0262	2.1167	2.1429	**45**
50	0.17811	0.01603	1703.2	1703.2	18.07	1065.6	1083.7	0.0361	2.0903	2.1264	**50**
60	0.2563	0.01604	1206.6	1206.7	28.06	1059.9	1088.0	0.0555	2.0393	2.0948	**60**
70	0.3631	0.01606	867.8	867.9	38.04	1054.3	1092.3	0.0745	1.9902	2.0647	**70**
80	0.5069	0.01608	633.1	633.1	48.02	1048.6	1096.6	0.0932	1.9428	2.0360	**80**
90	0.6982	0.01610	468.0	468.0	57.99	1042.9	1100.9	0.1115	1.8972	2.0087	**90**
100	0.9492	0.01613	350.3	350.4	67.97	1037.2	1105.2	0.1295	1.8531	1.9826	**100**
110	1.2748	0.01617	265.3	265.4	77.94	1031.6	1109.5	0.1471	1.8106	1.9577	**110**
120	1.6924	0.01620	203.25	203.27	87.92	1025.8	1113.7	0.1645	1.7694	1.9339	**120**
130	2.2225	0.01625	157.32	157.34	97.90	1020.0	1117.9	0.1816	1.7296	1.9112	**130**
140	2.8886	0.01629	122.99	123.01	107.89	1014.1	1122.0	0.1984	1.6910	1.8894	**140**
150	3.718	0.01634	98.06	97.07	117.89	1008.2	1126.1	0.2149	1.6537	1.8685	**150**
160	4.741	0.01639	77.27	77.29	127.89	1002.3	1130.2	0.2311	1.6174	1.8485	**160**
170	5.992	0.01645	62.04	62.06	137.90	996.3	1134.2	0.2472	1.5822	1.8293	**170**
180	7.510	0.01651	50.21	50.23	147.92	990.2	1138.1	0.2630	1.5480	1.8109	**180**
190	9.339	0.01657	40.94	40.96	157.95	984.1	1142.0	0.2785	1.5147	1.7932	**190**
200	11.526	0.01663	33.62	33.64	167.99	977.9	1145.9	0.2938	1.4824	1.7762	**200**
210	14.123	0.01670	27.80	27.82	178.05	971.6	1149.7	0.3090	1.4508	1.7598	**210**
212	14.696	0.01672	26.78	26.80	180.07	970.3	1150.4	0.3120	1.4446	1.7566	**212**
220	17.186	0.01677	23.13	23.15	188.13	965.2	1153.4	0.3239	1.4201	1.7440	**220**
230	20.780	0.01684	19.365	19.382	198.23	958.8	1157.0	0.3387	1.3901	1.7288	**230**
240	24.969	0.01692	16.306	16.323	208.34	952.2	1160.5	0.3531	1.3609	1.7140	**240**
250	29.825	0.01700	13.804	13.821	216.48	945.5	1164.0	0.3675	1.3323	1.6998	**250**
260	35.429	0.01709	11.746	11.763	228.64	938.7	1167.3	0.3817	1.3043	1.6860	**260**
270	41.858	0.01717	10.044	10.061	238.84	931.8	1170.6	0.3958	1.2769	1.6727	**270**
280	49.203	0.01726	8.628	8.645	249.06	924.7	1173.8	0.4096	1.2501	1.6597	**280**
290	57.556	0.01735	7.444	7.461	259.31	917.5	1176.8	0.4234	1.2238	1.6472	**290**
300	67.013	0.01745	6.449	6.466	269.59	910.1	1179.7	0.4369	1.1980	1.6350	**300**
310	77.68	0.01755	5.609	5.626	279.92	902.6	1182.5	0.4504	1.1727	1.6231	**310**
320	89.66	0.01765	4.896	4.914	290.28	894.9	1185.2	0.4637	1.1478	1.6115	**320**
330	103.06	0.01776	4.289	4.307	300.68	887.0	1187.7	0.4769	1.1233	1.6002	**330**
340	118.01	0.01787	3.770	3.788	311.13	879.0	1190.1	0.4900	1.0992	1.5891	**340**
350	134.63	0.01799	3.324	3.342	321.63	870.7	1192.3	0.5029	1.0754	1.5783	**350**
360	153.04	0.01811	2.939	2.957	332.18	862.2	1194.4	0.5158	1.0519	1.5677	**360**
370	173.37	0.01823	2.606	2.625	342.79	853.5	1196.3	0.5286	1.0287	1.5573	**370**
380	195.77	0.01836	2.317	2.335	353.45	844.6	1198.1	0.5413	1.0059	1.5471	**380**
390	220.37	0.01850	2.0651	2.0836	364.17	835.4	1199.6	0.5539	0.9832	1.5371	**390**
400	247.31	0.01864	1.8447	1.8633	374.97	826.0	1201.0	0.5664	0.9608	1.5272	**400**
410	276.75	0.01878	1.6512	1.6700	385.83	816.3	1202.1	0.5788	0.9386	1.5174	**410**
420	308.83	0.01894	1.4811	1.5000	396.77	806.3	1203.1	0.5912	0.9166	1.5078	**420**
430	343.72	0.01910	1.3308	1.3499	407.79	796.0	1203.8	0.6035	0.8947	1.4982	**430**
440	381.59	0.01926	1.1979	1.2171	418.90	785.4	1204.3	0.6158	0.8730	1.4887	**440**
450	422.6	0.0194	1.0799	1.0993	430.1	774.5	1204.6	0.6280	0.8513	1.4793	**450**
460	466.9	0.0196	0.9748	0.9944	441.4	763.2	1204.6	0.6402	0.8298	1.4700	**460**
470	514.7	0.0198	0.8811	0.9009	452.8	751.5	1204.3	0.6523	0.8083	1.4606	**470**
480	566.1	0.0200	0.7972	0.8172	464.4	739.4	1203.7	0.6645	0.7868	1.4513	**480**
490	621.4	0.0202	0.7221	0.7423	476.0	726.8	1202.8	0.6766	0.7653	1.4419	**490**
500	680.8	0.0204	0.6545	0.6749	487.8	713.9	1201.7	0.6887	0.7438	1.4325	**500**
520	812.4	0.0209	0.5385	0.5594	511.9	686.4	1198.2	0.7130	0.7006	1.4136	**520**
540	962.5	0.0215	0.4434	0.4649	536.6	656.6	1193.2	0.7374	0.6568	1.3942	**540**
550	1133.1	0.0221	0.3647	0.3868	562.2	624.2	1186.4	0.7621	0.6121	1.3742	**560**
580	1325.8	0.0228	0.2989	0.3217	588.9	588.4	1177.3	0.7872	0.5659	1.3532	**580**
600	1542.9	0.0236	0.2432	0.2668	617.0	548.5	1165.5	0.8131	0.5176	1.3307	**600**
620	1786.6	0.0247	0.1955	0.2201	646.7	503.6	1150.3	0.8398	0.4664	1.3062	**620**
640	2059.7	0.0260	0.1538	0.1798	678.6	452.0	1130.5	0.8679	0.4110	1.2789	**640**
650	2365.4	0.0278	0.1165	0.1442	714.2	390.2	1104.4	0.8987	0.3485	1.2472	**660**
630	2708.1	0.0305	0.0810	0.1115	757.3	309.9	1067.2	0.9351	0.2719	1.2071	**680**
700	3093.7	0.0369	0.0392	0.0761	823.3	172.1	995.4	0.9905	0.1484	1.1389	**700**
705.4	3206.2	0.0503	0	0.0503	902.7	0	902.7	1.0580	0	1.0580	**705.4**

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[a] Abridged from "Thermodynamic Properties of Steam," by Joseph H. Keenan and Frederick G. Keyes, John Wiley & Sons, Inc., New York, 1937.

TABLE 3. THERMODYNAMIC PROPERTIES OF STEAM

Superheated-steam table[a]

Abs. press., psia (sat. temp.)		Temp., F												
		200	**300**	**400**	**500**	**600**	**700**	**800**	**900**	**1000**	**1100**	**1200**	**1400**	**1600**
1 (101.74)	*v*...........	392.6	452.3	512.0	571.6	631.2	690.8	750.4	809.9	869.5	929.1	988.7	1107.8	1227.0
	h...........	1150.4	1195.8	1241.7	1288.3	1335.7	1383.8	1432.8	1482.7	1533.5	1585.2	1637.7	1745.7	1857.5
	s...........	2.0512	2.1153	2.1720	2.2233	2.2702	2.3137	2.3542	2.3923	2.4283	2.4625	2.4952	2.5566	2.6137
5 (162.24)	*v*...........	78.16	90.25	102.26	114.22	126.16	138.10	150.03	161.95	173.87	185.79	197.71	221.6	245.4
	h...........	1148.8	1195.0	1241.2	1288.0	1335.4	1383.6	1432.7	1482.6	1533.4	1585.1	1637.7	1745.7	1857.4
	s...........	1.8718	1.9370	1.9942	2.0456	2.0927	2.1361	2.1767	2.2148	2.2509	2.2851	2.3178	2.3792	2.4363
10 (193.21)	*v*...........	38.85	45.00	51.04	57.05	63.03	69.01	74.98	80.95	86.92	92.88	98.84	110.77	122.69
	h...........	1146.6	1193.9	1240.6	1287.5	1335.1	1383.4	1432.5	1482.4	1533.2	1585.0	1637.6	1745.6	1857.3
	s...........	1.7927	1.8595	1.9172	1.9689	2.0160	2.0596	2.1002	2.1383	2.1744	2.2086	2.2413	2.3028	2.3598
14.696 (212.00)	*v*...........	......	30.53	34.68	38.78	42.86	46.94	51.00	55.07	59.13	63.19	67.25	75.37	83.48
	h...........	......	1192.8	1239.9	1287.1	1334.8	1383.2	1432.3	1482.3	1533.1	1584.8	1637.5	1745.5	1857.3
	s...........	......	1.8160	1.8743	1.9261	1.9734	2.0170	2.0576	2.0958	2.1319	2.1662	2.1989	2.2603	2.3174
20 (227.96)	*v*...........	......	22.36	25.43	28.46	31.47	34.47	37.46	40.45	43.44	46.42	49.41	55.37	61.34
	h...........	......	1191.6	1239.2	1286.6	1334.4	1382.9	1432.1	1482.1	1533.0	1584.7	1637.4	1745.4	1857.2
	s...........	......	1.7808	1.8396	1.8918	1.9392	1.9829	2.0235	2.0618	2.0978	2.1321	2.1648	2.2263	2.2834
40 (267.25)	*v*...........	......	11.040	12.628	14.168	15.688	17.198	18.702	20.20	21.70	23.20	24.69	27.68	30.66
	h...........	......	1186.8	1236.5	1284.8	1333.1	1381.9	1431.3	1481.4	1532.4	1584.3	1637.0	1745.1	1857.0
	s...........	......	1.6994	1.7608	1.8140	1.8619	1.9058	1.9467	1.9850	2.0212	2.0555	2.0883	2.1498	2.2069
60 (292.71)	*v*...........	......	7.259	8.357	9.403	10.427	11.441	12.449	13.452	14.454	15.453	16.451	18.446	20.44
	h...........	......	1181.6	1233.6	1283.0	1331.8	1380.9	1430.5	1480.8	1531.9	1583.8	1636.6	1744.8	1856.7
	s...........	......	1.6492	1.7135	1.7678	1.8162	1.8605	1.9015	1.9400	1.9762	2.0106	2.0434	2.1049	2.1621
80 (312.03)	*v*...........	......	......	6.220	7.020	7.797	8.562	9.322	10.077	10.830	11.582	12.332	13.830	15.325
	h...........	......	......	1230.7	1281.1	1330.5	1379.9	1429.7	1480.1	1531.3	1583.4	1636.2	1744.5	1856.5
	s...........	......	......	1.6791	1.7346	1.7836	1.8281	1.8694	1.9079	1.9442	1.9787	2.0115	2.0731	2.1303
100 (327.81)	*v*...........	......	......	4.937	5.589	6.218	6.835	7.446	8.052	8.656	9.259	9.860	11.060	12.258
	h...........	......	......	1227.6	1279.1	1329.1	1378.9	1428.9	1479.5	1530.8	1582.9	1635.7	1744.2	1856.2
	s...........	......	......	1.6518	1.7085	1.7581	1.8029	1.8443	1.8829	1.9193	1.9538	1.9867	2.0484	2.1056
120 (341.25)	*v*...........	......	......	4.081	4.636	5.165	5.683	6.195	6.702	7.207	7.710	8.212	9.214	10.213
	h...........	......	......	1224.4	1277.2	1327.7	1377.8	1428.1	1478.8	1530.2	1582.4	1635.3	1743.9	1856.0
	s...........	......	......	1.6287	1.6869	1.7370	1.7822	1.8237	1.8625	1.8990	1.9335	1.9664	2.0281	2.0854
140 (353.02)	*v*...........	......	......	3.468	3.954	4.413	4.861	5.301	5.738	6.172	6.604	7.035	7.895	8.752
	h...........	......	......	1221.1	1275.2	1326.4	1376.8	1427.3	1478.2	1529.7	1581.9	1634.9	1743.5	1855.7
	s...........	......	......	1.6087	1.6683	1.7190	1.7645	1.8063	1.8451	1.8817	1.9163	1.9493	2.0110	2.0683
160 (363.53)	*v*...........	......	......	3.008	3.443	3.849	4.244	4.631	5.015	5.396	5.775	6.152	6.906	7.656
	h...........	......	......	1217.6	1273.1	1325.0	1375.7	1426.4	1477.5	1529.1	1581.4	1634.5	1743.2	1855.5
	s...........	......	......	1.5908	1.6519	1.7033	1.7491	1.7911	1.8301	1.8667	1.9014	1.9344	1.9962	2.0535
180 (373.06)	*v*...........	......	......	2.649	3.044	3.411	3.764	4.110	4.452	4.792	5.129	5.466	6.136	6.804
	h...........	......	......	1214.0	1271.0	1323.5	1374.7	1425.6	1476.8	1528.6	1581.0	1634.1	1742.9	1855.2
	s...........	......	......	1.5745	1.6373	1.6894	1.7355	1.7776	1.8167	1.8534	1.8882	1.9212	1.9831	2.0404
200 (381.79)	*v*...........	......	......	2.361	2.726	3.060	3.380	3.693	4.002	4.309	4.613	4.917	5.521	6.123
	h...........	......	......	1210.3	1268.9	1322.1	1373.6	1424.8	1476.2	1528.0	1580.5	1633.7	1742.6	1855.0
	s...........	......	......	1.5594	1.6240	1.6767	1.7232	1.7655	1.8048	1.8415	1.8763	1.9094	1.9713	2.0287
220 (389.86)	*v*...........	......	......	2.125	2.465	2.772	3.066	3.352	3.634	3.913	4.191	4.467	5.017	5.565
	h...........	......	......	1206.5	1266.7	1320.7	1372.0	1424.0	1475.5	1527.5	1580.0	1633.3	1742.3	1854.7
	s...........	......	......	1.5453	1.6117	1.6652	1.7120	1.7545	1.7939	1.8308	1.8656	1.8987	1.9607	2.0181
240 (397.37)	*v*...........	......	......	1.9276	2.247	2.533	2.804	3.068	3.327	3.584	3.839	4.093	4.597	5.100
	h...........	......	......	1202.5	1264.5	1319.2	1371.5	1423.2	1474.8	1526.9	1579.6	1632.9	1742.0	1854.5
	s...........	......	......	1.5219	1.6003	1.6546	1.7017	1.7444	1.7839	1.8209	1.8558	1.8889	1.9510	2.0084
260 (404.42)	*v*...........	......	......	......	2.063	2.330	2.582	2.827	3.067	3.305	3.541	3.776	4.242	4.707
	h...........	......	......	......	1262.3	1317.7	1370.4	1422.3	1474.2	1526.3	1579.1	1632.5	1741.7	1854.2
	s...........	......	......	......	1.5897	1.6447	1.6922	1.7352	1.7748	1.8118	1.8467	1.8799	1.9420	1.9995
280 (411.05)	*v*...........	......	......	......	1.9047	2.156	2.392	2.621	2.845	3.066	3.286	3.504	3.938	4.370
	h...........	......	......	......	1260.0	1316.2	1369.4	1421.5	1473.5	1525.8	1578.6	1632.1	1741.4	1854.0
	s...........	......	......	......	1.5796	1.6354	1.6834	1.7265	1.7662	1.8033	1.8383	1.8716	1.9337	1.9912
300 (417.33)	*v*...........	......	......	......	1.7675	2.005	2.227	2.442	2.652	2.859	3.065	3.269	3.674	4.078
	h...........	......	......	......	1257.6	1314.7	1368.3	1420.6	1472.8	1525.2	1578.1	1631.7	1741.0	1853.7
	s...........	......	......	......	1.5701	1.6268	1.6751	1.7184	1.7582	1.7954	1.8305	1.8638	1.9260	1.9835
350 (431.72)	*v*...........	......	......	......	1.4923	1.7036	1.8980	2.084	2.266	2.445	2.622	2.798	3.147	3.493
	h...........	......	......	......	1251.5	1310.9	1365.5	1418.5	1471.1	1523.8	1577.0	1630.7	1740.3	1853.1
	s...........	......	......	......	1.5481	1.6070	1.6563	1.7002	1.7403	1.7777	1.8130	1.8463	1.9086	1.9663
400 (444.59)	*v*...........	......	......	......	1.2851	1.4770	1.6508	1.8161	1.9767	2.134	2.290	2.445	2.751	3.055
	h...........	......	......	......	1245.1	1306.9	1362.7	1416.4	1469.4	1522.4	1575.8	1629.6	1739.5	1852.5
	s...........	......	......	......	1.5281	1.5894	1.6398	1.6842	1.7247	1.7623	1.7977	1.8311	1.8936	1.9513

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TABLE 3. THERMODYNAMIC PROPERTIES OF STEAM (*Continued*)

Superheated-steam table[a]

Abs. pres., psia (sat. temp.)		Temp., F													
		500	**550**	**600**	**620**	**640**	**660**	**680**	**700**	**800**	**900**	**1000**	**1200**	**1400**	**1600**
450 (456.28)	*v*	1.1231	1.2155	1.3005	1.3332	1.3652	1.3967	1.4278	1.4584	1.6074	1.7516	1.8928	2.170	2.443	2.714
	h	1238.4	1272.0	1302.8	1314.6	1326.2	1337.5	1348.8	1359.9	1414.3	1467.7	1521.0	1628.6	1738.7	1851.9
	s	1.5095	1.5437	1.5735	1.5845	1.5951	1.6054	1.6153	1.6250	1.6699	1.7108	1.7486	1.8177	1.8803	1.9381
500 (467.01)	*v*	0.9927	1.0800	1.1591	1.1893	1.2188	1.2478	1.2763	1.3044	1.4405	1.5715	1.6996	1.9504	2.197	2.442
	h	1231.3	1266.8	1298.6	1310.7	1322.6	1334.2	1345.7	1357.0	1412.1	1466.0	1519.6	1627.6	1737.9	1851.3
	s	1.4919	1.5280	1.5588	1.5701	1.5810	1.5915	1.6016	1.6115	1.6571	1.6982	1.7363	1.8056	1.8683	1.9262
550 (476.94)	*v*	0.8852	0.9686	1.0431	1.0714	1.0989	1.1259	1.1523	1.1783	1.3038	1.4241	1.5414	1.7706	1.9957	2.219
	h	1223.7	1261.2	1294.3	1306.8	1318.9	1330.8	1342.5	1354.0	1409.9	1464.3	1518.2	1626.6	1737.1	1850.6
	s	1.4751	1.5131	1.5451	1.5568	1.5680	1.5787	1.5890	1.5991	1.6452	1.6868	1.7250	1.7946	1.8575	1.9155
600 (486.21)	*v*	0.7947	0.8753	0.9463	0.9729	0.9988	1.0241	1.0489	1.0732	1.1899	1.3013	1.4096	1.6208	1.8279	2.033
	h	1215.7	1255.5	1289.9	1302.7	1315.2	1327.4	1339.3	1351.1	1407.7	1462.5	1516.7	1625.5	1736.3	1850.0
	s	1.4586	1.4990	1.5323	1.5443	1.5558	1.5667	1.5773	1.5875	1.6343	1.6762	1.7147	1.7846	1.8476	1.9056
700 (503.10)	*v*	......	0.7277	0.7934	0.8177	0.8411	0.8639	0.8860	0.9077	1.0108	1.1082	1.2024	1.3853	1.5641	1.7405
	h	......	1243.2	1280.6	1294.3	1307.5	1320.3	1332.8	1345.0	1403.2	1459.0	1515.9	1623.5	1734.8	1848.8
	s	......	1.4722	1.5084	1.5212	1.5333	1.5449	1.5559	1.5665	1.6147	1.6573	1.6963	1.7666	1.8299	1.8881
800 (518.23)	*v*	......	0.6154	0.6779	0.7006	0.7223	0.7433	0.7635	0.7833	0.8763	0.9633	1.0470	1.2088	1.3662	1.5214
	h	......	1229.8	1270.7	1285.4	1299.4	1312.9	1325.9	1338.6	1398.6	1455.4	1511.0	1621.4	1733.2	1847.5
	s	......	1.4467	1.4863	1.5000	1.5129	1.5250	1.5366	1.5476	1.5972	1.6407	1.6801	1.7510	1.8146	1.8729
900 (531.98)	*v*	......	0.5264	0.5873	0.6089	0.6294	0.6491	0.6680	0.6863	0.7716	0.8506	0.9262	1.0714	1.2124	1.3509
	h	......	1215.0	1260.1	1275.9	1290.9	1305.1	1318.8	1332.1	1393.9	1451.8	1508.1	1619.3	1731.6	1846.3
	s	......	1.4216	1.4653	1.4800	1.4938	1.5066	1.5187	1.5303	1.5814	1.6257	1.6656	1.7371	1.8009	1.8595
1000 (544.61)	*v*	......	0.4533	0.5140	0.5350	0.5546	0.5733	0.5912	0.6084	0.6878	0.7604	0.8294	0.9615	1.0893	1.2146
	h	......	1198.3	1248.8	1265.9	1281.9	1297.0	1311.4	1325.3	1389.2	1448.2	1505.1	1617.3	1730.0	1845.0
	s	......	1.3961	1.4450	1.4610	1.4757	1.4893	1.5021	1.5141	1.5670	1.6121	1.6525	1.7245	1.7886	1.8474
1100 (556.31)	*v*	......	......	0.4532	0.4738	0.4929	0.5110	0.5281	0.5445	0.6191	0.6866	0.7503	0.8716	0.9885	1.1031
	h	......	......	1236.7	1255.3	1272.4	1288.5	1303.7	1318.3	1384.3	1444.5	1502.2	1615.2	1728.4	1843.8
	s	......	......	1.4251	1.4425	1.4583	1.4728	1.4862	1.4989	1.5535	1.5995	1.6405	1.7130	1.7775	1.8363
1200 (567.22)	*v*	......	......	0.4016	0.4222	0.4410	0.4586	0.4752	0.4909	0.5617	0.6250	0.6843	0.7967	0.9046	1.0101
	h	......	......	1223.5	1243.9	1262.4	1279.6	1295.7	1311.0	1379.3	1440.7	1499.2	1613.1	1726.9	1842.5
	s	......	......	1.4052	1.4243	1.4413	1.4568	1.4710	1.4843	1.5409	1.5879	1.6293	1.7025	1.7672	1.8263
1400 (587.10)	*v*	......	......	0.3174	0.3390	0.3580	0.3753	0.3912	0.4062	0.4714	0.5281	0.5805	0.6789	0.7727	0.8640
	h	......	......	1193.0	1218.4	1240.4	1260.3	1278.5	1295.5	1369.1	1433.1	1493.2	1608 9	1723.7	1840.0
	s	......	......	1.3639	1.3877	1.4079	1.4258	1.4419	1.4567	1.5177	1.5666	1.6093	1.6836	1.7489	1.8083
1600 (604.90)	*v*	......	......	......	0.2733	0.2936	0.3112	0.3271	0.3417	0.4034	0.4553	0.5027	0.5906	0.6738	0.7545
	h	......	......	......	1187.8	1215.2	1238.7	1259.6	1278.7	1358.4	1425.3	1487.0	1604.6	1720.5	1837.5
	s	......	......	......	1.3489	1.3741	1.3952	1.4137	1.4303	1.4964	1.5476	1.5914	1.6669	1.7328	1.7926
1800 (621.03)	*v*	......	......	......	......	0.2407	0.2597	0.2760	0.2907	0.3502	0.3986	0.4421	0.5218	0.5968	0.6693
	h	......	......	......	......	1185.1	1214.0	1238.5	1260.3	1347.2	1417.4	1489.8	1600.4	1717.3	1835.0
	s	......	......	......	......	1.3377	1.3638	1.3855	1.4044	1.4765	1.5301	1.5752	1.6520	1.7185	1.7786
2000 (635.82)	*v*	......	......	......	......	0.1936	0.2161	0.2337	0.2489	0.3074	0.3532	0.3935	0.4668	0.5352	0.6011
	h	......	......	......	......	1145.6	1184.9	1214.8	1240.0	1335.5	1409.2	1474.5	1596.1	1714.1	1832.5
	s	......	......	......	......	1.2945	1.3300	1.3564	1.3783	1.4576	1.5139	1.5603	1.6384	1.7055	1.7660
2500 (668.13)	*v*	......	......	......	......	......	......	0.1484	0.1686	0.2294	0.2710	0.3061	0.3678	0.4244	0.4784
	h	......	......	......	......	......	......	1132.3	1176.8	1303.6	1387.8	1458.4	1585.3	1706.1	1826.2
	s	......	......	......	......	......	......	1.2687	1.3073	1.4127	1.4772	1.5273	1.6088	1.6775	1.7389
3000 (695.36)	*v*	......	......	......	......	......	......	......	0.0984	0.1760	0.2159	0.2476	0.3018	0.3505	0.3966
	h	......	......	......	......	......	......	......	1060.7	1267.2	1365.0	1441.8	1574.3	1698.0	1819.9
	s	......	......	......	......	......	......	......	1.1966	1.3690	1.4439	1.4984	1.5837	1.6540	1.7163
3206.2 (705.40)	*v*	......	......	......	......	......	......	......	......	0.1583	0.1981	0.2288	0.2806	0.3267	0.3703
	h	......	......	......	......	......	......	......	......	1250.5	1355.2	1434.7	1569.8	1694.6	1817.2
	s	......	......	......	......	......	......	......	......	1.3508	1.4309	1.4874	1.5742	1.6452	1.7080
3500	*v*	......	......	......	......	......	......	......	0.0306	0.1364	0.1762	0.2058	0.2546	0.2977	0.3381
	h	......	......	......	......	......	......	......	780.5	1224.9	1340.7	1424.5	1563.3	1689.8	1813.6
	s	......	......	......	......	......	......	......	0.9515	1.3241	1.4127	1.4723	1.5615	1.6336	1.6968
4000	*v*	......	......	......	......	......	......	......	0.0287	0.1052	0.1462	0.1743	0.2192	0.2581	0.2943
	h	......	......	......	......	......	......	......	763.8	1174.8	1314.4	1406.8	1552.1	1681.7	1807.2
	s	......	......	......	......	......	......	......	0.9347	1.2757	1.3827	1.4482	1.5417	1.6154	1.6795
4500	*v*	......	......	......	......	......	......	......	0.0276	0.0798	0.1226	0.1500	0.1917	0.2273	0.2602
	h	......	......	......	......	......	......	......	753.5	1113.9	1286.5	1388.4	1540.8	1673.5	1800.9
	s	......	......	......	......	......	......	......	0.9235	1.2204	1.3529	1.4253	1.5235	1.5990	1.6640
5000	*v*	......	......	......	......	......	......	......	0.0268	0.0593	0.1036	0.1303	0.1696	0.2027	0.2329
	h	......	......	......	......	......	......	......	746.4	1047.1	1256.5	1369.5	1529.5	1665.3	1794.5
	s	......	......	......	......	......	......	......	0.9152	1.1622	1.3231	1.4034	1.5066	1.5839	1.6499
5500	*v*	......	......	......	......	......	......	......	0.0262	0.0463	0.0880	0.1143	0.1516	0.1825	0.2106
	h	......	......	......	......	......	......	......	741.3	985.0	1224.1	1349.3	1518.2	1657.0	1788.1
	s	......	......	......	......	......	......	......	0.9090	1.1093	1.2930	1.3821	1.4908	1.5699	1.6369

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[a] Abridged from "Thermodynamic Properties of Steam," by Joseph H. Keenan and Frederick G. Keyes, John Wiley & Sons, Inc., New York, 1937.

TABLE 4. PROPERTIES OF STEAM AT HIGH PRESSURES

Abridged interim steam table[a]

Pressure, psia		Temperature, °F																
		32	100	200	300	400	500	600	700	800	900	1000	1100	1200	1300	1400	1500	1600
5,500	*v*	0.01573	0.01588	0.01635	0.01708	0.01810	0.01955	0.02177	0.0262	0.0463	0.0880	0.1143	0.1343	0.1516	0.1674	0.1825	0.1968	0.2106
	h	16.18	82.38	180.5	280.0	381.8	488.3	603.9	741.3	985.0	1224.1	1349.3	1440.8	1518.2	1588.0	1657.0	1722.7	1788.1
	s	0.00016	0.12609	0.28735	0.42774	0.55352	0.67061	0 78510	0.9090	1.1093	1.2930	1.3821	1.4427	1.4908	1.5306	1.5699	1.6035	1.6369
6,000	*v*	0.01570	0.01585	0.01633	0.01705	0.01805	0.01947	0.02160	0.0257	0.0402	0.0755	0.1013	0.1207	0.1370	0.1518	0.1660	0.1794	0.1924
	h	17.61	83.66	181.6	280.9	382.5	488.6	603.1	737.2	944.9	1192.5	1330.5	1427.2	1507.7	1578.4	1648.1	1715.2	1781.8
	s	0.00010	0.12576	0.28681	0.42695	0.55241	0.66904	0.78249	0.9032	1.0744	1.2642	1.3622	1.4263	1.4765	1.5168	1.5564	1.5908	1.6247
6,500	*v*	0.01568	0.01583	0.01630	0.01702	0.01801	0.01941	0.02146	0.0252	0.0366	0.0656	0.0906	0.1093	0.1249	0.1390	0.1524	0.1650	0.1771
	h	19.03	84.94	182.8	281.9	383.2	489.0	602.5	733.6	918.2	1162.8	1310.5	1413.0	1496.8	1569.0	1639.8	1708.0	1775.6
	s	0.00004	0.12543	0.28627	0.42617	0.55132	0.66752	0.78004	0.8978	1.0502	1.2375	1.3425	1.4104	1.4626	1.5038	1.5441	1.5790	1.6134
7,000	*v*	0.01566	0.01581	0.01628	0.01699	0.01798	0.01935	0.02133	0.0248	0.0339	0.0577	0.0815	0.0997	0.1147	0.1282	0.1408	0.1528	0.1643
	h	20.45	86.22	183.9	282.9	383.9	489.4	602.0	730.4	899.4	1132.0	1290.4	1398.7	1485.8	1559.8	1631.7	1700.8	1769.4
	s	−0.00002	0.12510	0.28574	0.42540	0.55024	0.66604	0.77771	0.8930	1.0328	1.2105	1.3233	1.3951	1.4494	1.4916	1.5324	1.5679	1.6027
7,500	*v*	0.01563	0.01579	0.01626	0.01696	0.01794	0.01930	0.02122	0.0245	0.0319	0.0513	0.0738	0.0915	0.1060	0.1189	0.1310	0.1422	0.1532
	h	21.87	87.50	185.1	283.9	384.7	489.8	601.6	727.1	885.3	1101.1	1270.4	1384.3	1474.9	1550.7	1623.8	1693.6	1763.2
	s	−0.00008	0.12478	0.28520	0.42464	0.54918	0.66460	0.77545	0.8885	1.0194	1.1842	1.3046	1.3802	1.4366	1.4799	1.5214	1.5573	1.5926
8,000	*v*	0.01561	0.01577	0.01623	0.01693	0.01790	0.01924	0.02112	0.0242	0.0304	0.0458	0.0673	0.0844	0.0985	0.1108	0.1224	0.1331	0.1435
	h	23.27	88.78	186.2	284.9	385.4	490.2	601.3	724.6	875.9	1074.4	1250.8	1369.8	1463.9	1541.5	1616.0	1686.7	1757.1
	s	−0.00017	0.12446	0.28467	0.42388	0.54812	0.66317	0.77328	0.8845	1.0099	1.1614	1.2867	1.3657	1.4243	1.4686	1.5109	1.5472	1.5830
8,500	*v*	0.01558	0.01574	0.01620	0.01689	0.01787	0.01919	0.02103	0.0240	0.0293	0.0420	0.0617	0.0781	0.0918	0.1037	0.1148	0.1252	0.1350
	h	24.67	90.06	187.4	285.9	386.2	490.6	601.0	722.7	867.8	1052.6	1232.1	1355.2	1453.0	1532.4	1608.2	1679.9	1751.2
	s	−0.00024	0.12414	0.28414	0.42313	0.54706	0.66176	0.77117	0.8806	1.0006	1.1424	1.2698	1.3515	1.4124	1.4579	1.5009	1.5377	1.5739
9,000	*v*	0.01556	0.01572	0.01618	0.01686	0.01783	0.01914	0.02095	0.0238	0.0284	0.0393	0.0568	0.0726	0.0860	0.0974	0.1081	0.1181	0.1276
	h	26.07	91.33	188.5	286.9	386.9	491.0	600.8	721.0	861.8	1036.3	1213.2	1340.8	1442.0	1523.7	1600.5	1673.0	1745.6
	s	−0.00034	0.12381	0.28362	0.42238	0.54602	0.66036	0.76909	0.8775	0.9939	1.1271	1.2531	1.3378	1.4008	1.4476	1.4911	1.5284	1.5652
9,500	*v*	0.01554	0.01570	0.01616	0.01684	0.01780	0.01909	0.02088	0.0236	0.0279	0.0371	0.0527	0.0677	0.0807	0.0920	0.1022	0.1118	0.1209
	h	27.47	92.61	189.7	287.9	387.7	491.5	600.6	719.7	857.3	1023.8	1196.0	1326.7	1430.9	1515.4	1592.8	1666.1	1740.1
	s	−0.00042	0.12349	0.28309	0.42164	0.54499	0.65897	0.76709	0.8745	0.9886	1.1156	1.2379	1.3246	1.3895	1.4379	1.4818	1.5195	1.5570
10,000	*v*	0.01552	0.01568	0.01614	0.01682	0.01776	0.01904	0.02080	0.0234	0.0275	0.0353	0.0495	0.0635	0.0760	0.0872	0.0969	0.1061	0.1149
	h	28.84	93.88	190.8	288.9	388.5	491.9	600.4	718.4	853.3	1012.9	1181.0	1313.3	1419.7	1507.4	1585.5	1659.3	1734.7
	s	−0.00054	0.12317	0.28257	0.42089	0.54396	0.65759	0.76517	0.8714	0.9832	1.1050	1.2243	1.3122	1.3784	1.4286	1.4729	1.5108	1.5491

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[a] The complete Interim Steam Table prepared by the Subcommittee on Interim Steam Tables of the ASME Power Division is available from the Order Department of the ASME.

TABLE 5. THERMODYNAMIC PROPERTIES OF AIR (TEMPERATURE)[a]

T	h	p_r	u	v_r	ϕ	T	h	p_r	u	v_r	ϕ
400	95.53	0.4858	68.11	305.0	0.52890	890	213.80	8.081	152.80	40.80	0.72163
410	97.93	0.5295	69.82	286.8	0.53481	900	216.26	8.411	154.57	39.64	0.72438
420	100.32	0.5760	71.52	270.1	0.54058	910	218.72	8.752	156.34	38.52	0.72710
430	102.71	0.6253	73.23	254.7	0.54621	920	221.18	9.102	158.12	37.44	0.72979
440	105.11	0.6776	74.93	240.6	0.55172	930	223.64	9.463	159.89	36.41	0.73245
450	107.50	0.7329	76.65	227.45	0.55710	940	226.11	9.834	161.68	35.41	0.73509
460	109.90	0.7913	78.36	215.33	0.56235	950	228.58	10.216	163.46	34.45	0.73771
470	112.30	0.8531	80.07	204.08	0.56751	960	231.06	10.610	165.26	33.52	0.74030
480	114.69	0.9182	81.77	193.65	0.57255	970	233.53	11.014	167.05	32.63	0.74287
490	117.08	0.9868	83.49	183.94	0.57749	980	236.02	11.430	168.83	31.76	0.74540
500	119.48	1.0590	85.20	174.90	0.58233	990	238.50	11.858	170.63	30.92	0.74792
510	121.87	1.1349	86.92	166.46	0.58707	1000	240.98	12.298	172.43	30.12	0.75042
520	124.27	1.2147	88.62	158.58	0.59173	1010	243.48	12.751	174.24	29.34	0.75290
530	126.66	1.2983	90.34	151.22	0.59630	1020	245.97	13.215	176.04	28.59	0.75536
540	129.06	1.3860	92.04	144.32	0.60078	1030	248.45	13.692	177.84	27.87	0.75778
550	131.46	1.4779	93.76	137.85	0.60518	1040	250.95	14.182	179.66	27.17	0.76019
560	133.86	1.5742	95.47	131.78	0.60950	1050	253.45	14.686	181.47	26.48	0.76259
570	136.26	1.6748	97.19	126.08	0.61376	1060	255.96	15.203	183.29	25.82	0.76496
580	138.66	1.7800	98.90	120.70	0.61793	1070	258.47	15.734	185.10	25.19	0.76732
590	141.06	1.8899	100.62	115.65	0.62204	1080	260.97	16.278	186.93	24.58	0.76964
600	143.47	2.005	102.34	110.88	0.62607	1090	263.48	16.838	188.75	23.98	0.77196
610	145.88	2.124	104.06	106.38	0.63005	1100	265.99	17.413	190.58	23.40	0.77426
620	148.28	2.249	105.78	102.12	0.63395	1110	268.52	18.000	192.41	22.84	0.77654
630	150.68	2.379	107.50	98.11	0.63781	1120	271.03	18.604	194.25	22.30	0.77880
640	153.09	2.514	109.21	94.30	0.64159	1130	273.56	19.223	196.09	21.78	0.78104
650	155.50	2.655	110.94	90.69	0.64533	1140	276.08	19.858	197.94	21.27	0.78326
660	157.92	2.801	112.67	87.27	0.64902	1150	278.61	20.51	199.78	20.771	0.78548
670	160.33	2.953	114.40	84.03	0.65263	1160	281.14	21.18	201.63	20.293	0.78767
680	162.73	3.111	116.12	80.96	0.65621	1170	283.68	21.86	203.49	19.828	0.78985
690	165.15	3.276	117.85	78.03	0.65973	1180	286.21	22.56	205.33	19.377	0.79201
700	167.56	3.446	119.58	75.25	0.66321	1190	288.76	23.28	207.19	18.940	0.79415
710	169.98	3.623	121.32	72.60	0.66664	1200	291.30	24.01	209.05	18.514	0.79628
720	172.39	3.806	123.04	70.07	0.67002	1210	293.86	24.76	210.92	18.102	0.79840
730	174.82	3.996	124.78	67.67	0.67335	1220	296.41	25.53	212.78	17.700	0.80050
740	177.23	4.193	126.51	65.38	0.67665	1230	298.96	26.32	214.65	17.311	0.80258
750	179.66	4.396	128.25	63.20	0.67991	1240	301.52	27.13	216.53	16.932	0.80466
760	182.08	4.607	129.99	61.10	0.68312	1250	304.08	27.96	218.40	16.563	0.80672
770	184.51	4.826	131.73	59.11	0.68629	1260	306.65	28.80	220.28	16.205	0.80876
780	186.94	5.051	133.47	57.20	0.68942	1270	309.22	29.67	222.16	15.857	0.81079
790	189.38	5.285	135.22	55.38	0.69251	1280	311.79	30.55	224.05	15.518	0.81280
800	191.81	5.526	136.97	53.63	0.69558	1290	314.36	31.46	225.93	15.189	0.81481
810	194.25	5.775	138.72	51.96	0.69860	1300	316.94	32.39	227.83	14.868	0.81680
820	196.69	6.033	140.47	50.35	0.70160	1310	319.53	33.34	229.73	14.557	0.81878
830	199.12	6.299	142.22	48.81	0.70455	1320	322.11	34.31	231.63	14.253	0.82075
840	201.56	6.573	143.98	47.34	0.70747	1330	324.69	35.30	233.52	13.958	0.82270
850	204.01	6.856	145.74	45.92	0.71037	1340	327.29	36.31	235.43	13.670	0.82464
860	206.46	7.149	147.50	44.57	0.71323	1350	329.88	37.35	237.34	13.391	0.82658
870	208.90	7.450	149.27	43.26	0.71606	1360	332.48	38.41	239.25	13.118	0.82848
880	211.35	7.761	151.02	42.01	0.71886	1370	335.09	39.49	241.17	12.851	0.83039

TABLE 5. THERMODYNAMIC PROPERTIES OF AIR (TEMPERATURE)[a] (*Continued*)

T	h	p_r	u	v_r	ϕ	T	h	p_r	u	v_r	ϕ
1380	337.68	40.59	243.08	12.593	0.83229	2500	645.78	435.7	474.40	2.125	0.99497
1390	340.29	41.73	245.00	12.340	0.83417	2600	674.49	513.5	496.26	1.8756	1.00623
1400	342.90	42.88	246.93	12.095	0.83604	2700	703.35	601.9	518.26	1.6617	1.01712
1410	345.52	44.06	248.86	11.855	0.83790	2800	732.33	702.0	540.40	1.4775	1.02767
1420	348.14	45.26	250.79	11.622	0.83975	2900	761.45	814.8	562.66	1.3184	1.03788
1430	350.75	46.49	252.72	11.394	0.84158	3000	790.68	941.4	585.04	1.1803	1.04779
1440	353.37	47.75	254.66	11.172	0.84341	3100	820.03	1083.4	607.53	1.0600	1.05741
1450	356.00	49.03	256.60	10.954	0.84523	3200	849.48	1241.7	630.12	0.9546	1.06676
1460	358.63	50.34	258.54	10.743	0.84704	3300	879.02	1418.0	652.81	0.8621	1.07585
1470	361.27	51.68	260.49	10.537	0.84884	3400	908.66	1613.2	675.60	0.7807	1.08470
1480	363.89	53.04	262.44	10.336	0.85062	3500	938.40	1829.3	698.48	0.7087	1.09332
1490	366.53	54.43	264.38	10.140	0.85239	3600	968.21	2067.9	721.44	0.6449	1.10172
1500	369.17	55.86	266.34	9.948	0.85416	3700	998.11	2330.3	744.48	0.5882	1.10991
1550	382.42	63.40	276.17	9.056	0.86285	3800	1028.09	2618.4	767.60	0.5376	1.11791
1600	395.74	71.73	286.06	8.263	0.87130	3900	1058.14	2934.4	790.80	0.4923	1.12571
1650	409.13	80.89	296.03	7.556	0.87954	4000	1088.26	3280	814.05	0.4518	1.13334
1700	422.59	90.95	306.06	6.924	0.88758	4100	1118.46	3656	837.40	0.4154	1.14079
1750	436.12	101.98	316.16	6.357	0.89542	4200	1148.72	4067	860.81	0.3826	1.14809
1800	449.71	114.03	326.32	5.847	0.90308	4300	1179.04	4513	884.28	0.3529	1.15522
1850	463.37	127.18	336.55	5.388	0.91056	4400	1209.42	4997	907.81	0.3262	1.16221
1900	477.09	141.51	346.85	4.974	0.91788	4500	1239.86	5521	931.39	0.3019	1.16905
1950	490.88	157.10	357.20	4.598	0.92504	4600	1270.36	6089	955.04	0.2799	1.17575
2000	504.71	174.00	367.61	4.258	0.93205	4700	1300.92	6701	978.73	0.2598	1.18232
2100	532.55	212.1	388.60	3.667	0.94564	4800	1331.51	7362	1002.48	0.2415	1.18876
2200	560.59	256.6	409.78	3.176	0.95868	4900	1362.17	8073	1026.28	0.2248	1.19508
2300	588.82	308.1	431.16	2.765	0.97123	5000	1392.87	8837	1050.12	0.2096	1.20129
2400	617.22	367.6	452.70	2.419	0.98331						

[a] Reprinted with permission from Joseph H. Keenan and Joseph Kaye, Gas Tables, "Thermodynamic Properties of Air, Products of Combustion and Component Gases," Copyright, 1948, John Wiley & Sons, Inc.

Symbols and units:

T = absolute temperature, °R
h = enthalpy, Btu per lb
p_r = relative pressure
u = internal energy, Btu per lb
v_r = relative volume
ϕ = entropy at constant pressure = $\int c_p(dT/T)$

Procedure for determining pressure, volumes, and changes in entropy of a process:

For an arbitrary assigned pressure and temperature datum, the absolute pressure that corresponds to any temperature listed in the table for an isentropic condition may be computed from the corresponding relative pressures.

For example, if the datum is that of standard conditions of 14.696 psia and 520 R, the pressure p_b, corresponding to a temperature, say 800 R, is

$$p_b = p_a \frac{p_{rb}}{p_{ra}} = 14.696 \times \frac{5.526}{1.2147} = 66.86 \text{ psia}$$

In a similar manner, the specific volume at 800 R is

$$v_b = v_a \frac{v_{rb}}{v_{ra}} = \frac{RT_a}{144 p_a} \times \frac{v_{rb}}{v_{ra}}$$
$$= \frac{53.37 \times 520}{144 \times 14.696} \times \frac{53.63}{158.58} = 4.436 \text{ cu ft}$$

The change of entropy between any two temperatures also involving changes in pressure may be computed thus:

$$s_2 - s_1 = (\phi_2 - \phi_1) - \frac{R}{J} \ln \frac{p_2}{p_1}$$

Table 6. Properties of Refractories[a]

Type	Melting point, °F	Normal-use limit, °F	Bulk density, lb per cu ft
Refractory brick:			
Alumina-silica			
Fire-clay base			
First quality	3090–3175	2400–2700	125–140
90% alumina	3390–3425	2800–3200	172
Kaolin base	3190	2800–2900	135–145
Mullite base	3325–3350	2900–3200	150–160
Kyanite base	3250–3300	2800–3000	140–150
Silica (standard)	3142	3000	100–105
Magnesite (burned)	5070	3000–4000	160–165
Chrome (burned)	3540–3990	2800–3200	185–190
Forsterite	3461	3000	150–155
Silicon carbide	Dissociates 4080	2800–3200	155
Fused alumina	3722	3400	175–195
Zircon	4532	3400	205
Zirconia (stabilized)	4870	4300	275
Carbon (graphite)	6330	4000	137
Insulating firebrick:			
ASTM Group 16	....	1600	21–37
ASTM Group 20	....	2000	26–45
ASTM Group 23	....	2300	27–47
ASTM Group 26	....	2600	43–64
ASTM Group 28	....	2800	45–65
Others	....	2900–3250	52–81

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[a] Courtesy of C. L. Norton, Refractories for Every Use, abridged, *Chemical Engineering*, June, 1953.

Table 7. Thermal Conductivities of Insulating Materials[a]
(k = Btu per hr-sq ft-°F-ft)

Material	Density, lb per cu ft	Temperature, °F			
		32	70	212	500
85 per cent magnesia pipe covering	11	.....	0.034	0.038	0.048
Asbestos fiber, standard, pipe insulation and block	14	.....	0.027	0.033	0.050
Diatomaceous-earth pipe covering	23	.....	0.040	0.045	0.053
Mineral wool, molded pipe covering	18	.....	0.033	0.035	0.035
Cellular glass, pipe insulation	10	0.027	0.035		
Air cell, pipe covering	9	.....	0.041	0.046	
Asbestos, laminated felts, pipe insulation and blocks	23	.....	0.033	0.038	0.047

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[a] Courtesy of L. S. Marks, "Mechanical Engineers' Handbook," 5th ed., McGraw-Hill Book Company, Inc., New York, 1951; R. Thomas, Insulation for Heat and Cold, abridged, *Chemical Engineering*, June, 1953.

Table 8. Thermal Conductivities of Gases (Low Pressures)[a]
(k = Btu per hr-sq ft-°F-ft)

Temp, °F	Gas				
	Air	Nitrogen	Oxygen	Carbon dioxide	Steam
80	0.015	0.015	0.015	0.010	
260	0.020	0.019	0.020	0.014	0.015
440	0.023	0.023	0.024	0.019	0.020
620	0.027	0.027	0.028	0.024	0.024
800	0.030	0.030	.....	0.029	0.029
980	0.033	0.033	.....	0.032	0.034
1340	0.039	0.035	.....	0.039	
1700	.....	0.042	.....	0.045	
1880	.....	.....	.....	0.048	

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[a] Courtesy of J. Hilsenrath and Y. S. Touloukian, The Viscosity, Thermal Conductivity and Prandtl Numbers for Air, O_2, N_2, NO, H_2, CO, CO_2, H_2O, He and A, *Transactions of the ASME*, 1954.

TABLE 9. THERMAL CONDUCTIVITIES OF MISCELLANEOUS SUBSTANCES[a]

(k = Btu per hr-sq ft-°F-ft)

Material	Temp, °F	k
Metals:		
Aluminum: 14S alloy (4.4% Cu, 0.8% Si, 0.75% Mn, 0.35% Mg)	200	113
	400	111
	600	107
Aluminum: 355 alloy (1.2% Cu, 5.0% Si, 0.5% Mg)	200	95
	400	97
	600	97
Aluminum	64	117
	212	119
(Data for aluminum vary considerably owing to composition and treatment)		
Brass (61.5% Cu, 35.5% Zn, 3.0% Pb)	200	67
	400	73.5
	600	80.5
Copper	64	224
	212	218
Inconel X	200	9.1
	400	10.4
	600	11.7
	800	13.0
	1000	14.3
Lead	64	36
	212	34
Magnesium	32–212	92
Nickel	64	36
	212	34
Tin	64	36
	212	34
Zinc	64	65
	212	64
Tube and Plate Alloys:		
Red brass (85.0% Cu, 15.0% Zn)	68	92
Muntz metal (60.0% Cu, 40.0% Zn)	68	71
Admiralty metal (71.0% Cu, 28.0% Zn, 1.0% Sn, 0.05% As)	68	64
Naval brass (60.0% Cu, 39.25% Zn, 0.75% Sn)	68	67
Manganese bronze (58.5% Cu, 39.2% Zn, 1.0% Fe, 1.0% Sn, 0.3% Mn)	68	61
Aluminum brass (Revalon) (76.0% Cu, 22.0% Zn, 2.0% Al, 0.05% As)	68	58
Aluminum bronze (95.0% Cu, 5.0% Al, 0.35% As)	68	48
Cupronickel (88.5% Cu, 10.0% Ni, 1.5% Fe)	68	27
Cupronickel (70.0% Cu, 30.0% Ni)	68	17

TABLE 9. THERMAL CONDUCTIVITIES OF MISCELLANEOUS SUBSTANCES[a]

(*Continued*)

(k = Btu per hr-sq ft-°F-ft)

Material	Temp, °F	k
Various solids:		
Scale, boiler:		
Silicate	570	0.05–0.1
Sulfate	570	0.34–1.34
Cement	......	0.17
Concrete:		
Cinder	......	0.167–0.42
Stone	......	0.5–0.75
Corkboard	86	0.025
Glass	......	0.333–0.5
Limestone	......	0.3–0.75
Marble	......	1.2–1.7
Paper	......	0.075
Plaster	......	0.25–0.5
Rubber:		
Hard	100	0.092
Soft	86	0.08
Sandstone	104	1.1
Slate	201	0.86
Woods (across grain):		
Balsa	86	0.03–0.048
Cypress	86	0.056
Oak	......	0.12
White pine	59	0.087
Yellow pine	......	0.085
Liquids:		
Gasoline	86	0.078
Kerosene	68	0.086
Mercury	82	4.83
Petroleum oils	86–212	0.08
Sodium	410	46
Water	32	0.343
	100	0.363
	200	0.393
	300	0.395
	420	0.376
	620	0.275

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[a] Courtesy of L. S. Marks, "Mechanical Engineers' Handbook," 5th ed., McGraw-Hill Book Company, Inc., New York, 1951; A. Schack, "Industrial Heat Transfer," John Wiley & Sons, Inc., 1933; J. E. Evans, Jr., Thermal Conductivity of Several Metals and Alloys, "Product Engineering Handbook for 1955," "Technical Information on Revere Copper and Copper Alloys," Revere Copper and Brass, Inc.; W. H. McAdams, "Heat Transmission," 3d ed., McGraw-Hill Book Company, Inc., New York, 1954.

TABLE 10. VISCOSITY OF LIQUIDS (LOW PRESSURES)[a]
(μ = lb per hr-ft)

Temp, °F	Liquid						
	Water	No. 6 fuel oil	No. 5 fuel oil	No. 2 fuel oil	Kerosene	SAE 30 oil	SAE 10 oil
60	2.735	. . .	370	8.8	4.0	. . .	340
80	2.178	. . .	170	6.7	3.3	480	170
100	1.725	. . .	81	5.3	2.8	220	88
120	1.452	702	48	4.2	2.4	120	55
140	1.185	320	30	3.4	2.1	70	36
160	0.967	150	20	2.8	1.8	47	29
180	0.823	83	15	. . .	. . .	32	18
200	0.690	50	11	. . .	. . .	23	14
220	0.570	34	7.7	. . .	. . .	18	11
240	0.484	20	6.9	. . .	. . .	13	8.8
260	0.412	14					
300	0.314						
320	0.266						

[a] Courtesy of W. H. McAdams, "Heat Transmission," 3d ed., McGraw-Hill Book Company, Inc., New York, 1954, and various commercial literature.

TABLE 11. VISCOSITY OF GASES (LOW PRESSURES)[a]
(μ = lb per hr-ft)

Temp, °F	Gas				
	Air	Nitrogen	Oxygen	Carbon dioxide	Steam
80	0.045	0.043	0.050	0.036	0.024
260	0.055	0.053	0.062	0.047	0.033
440	0.065	0.062	0.072	0.056	0.041
620	0.073	0.071	0.082	0.065	0.050
800	0.081	0.078	0.091	0.073	0.059
980	0.088	0.085	. . . . .	0.080	0.067
1340	0.101	0.099	. . . . .	0.094	
1700	0.112	0.112	. . . . .	0.107	
1880	0.117	0.117	. . . . .	0.113	

[a] Courtesy of J. Hilsenrath and Y. S. Touloukian, The Viscosity, Thermal Conductivity and Prandtl Numbers for Air, O_2, N_2, NO, H_2, CO, CO_2, H_2O, He and A, *Transactions of the ASME*, 1954.

TABLE 12. TUBE DIMENSIONS BY BIRMINGHAM WIRE GAUGE (BWG)

OD, in.	Thickness		ID, in.
	Bwg	In.	
¼ (0.250)	22	0.028	0.194
	24	0.022	0.206
	26	0.018	0.214
⅜ (0.375)	18	0.049	0.277
	20	0.035	0.305
	22	0.028	0.319
½ (0.500)	18	0.049	0.402
	20	0.035	0.430
⅝ (0.625)	10	0.134	0.357
	12	0.109	0.407
	14	0.083	0.458
	16	0.065	0.495
¾ (0.750)	10	0.134	0.482
	12	0.109	0.532
	14	0.083	0.584
	16	0.065	0.620
	18	0.049	0.652
⅞ (0.875)	10	0.134	0.607
	14	0.083	0.709
	16	0.065	0.745
	18	0.049	0.777
	20	0.035	0.805
1.0 (1.000)	10	0.134	0.732
	12	0.109	0.782
	14	0.083	0.834
	16	0.065	0.870
	18	0.049	0.902
1¼ (1.250)	10	0.134	0.982
	12	0.109	1.032
	14	0.083	1.084
	16	0.065	1.120
	18	0.049	1.152
1½ (1.500)	10	0.134	1.232
	12	0.109	1.282
	14	0.083	1.334
	16	0.065	1.370
2 (2.000)	11	0.120	1.760
	13	0.095	1.810
2½ (2.500)	9	0.148	2.200

Table 13. Standard-weight Iron-pipe and -tube Dimensions

Nominal size, in.	Diameter, in.		Thickness, in.
	OD	ID	
⅛	0.405	0.269	0.068
¼	0.540	0.364	0.088
⅜	0.675	0.493	0.091
½	0.840	0.622	0.109
¾	1.050	0.824	0.113
1	1.315	1.049	0.133
1¼	1.660	1.380	0.140
1½	1.900	1.610	0.145
2	2.375	2.067	0.154
2½	2.875	2.469	0.203
3	3.500	3.068	0.216
3½	4.000	3.548	0.226
4	4.500	4.026	0.237
5	5.563	5.047	0.258
6	6.625	6.065	0.280
8	8.625	7.981	0.322
10	10.750	10.020	0.365
12	12.750	12.000	0.375
14 OD	14.000	13.250	0.375
16 OD	16.000	15.250	0.375
18 OD	18.000	17.182	0.409

Table 14. Values of $(1 + i)^n$ for Given Values of i and n

i n	0.01	0.02	0.03	0.04	0.05	0.06	0.07	0.08	0.09	0.10
1	1.01	1.02	1.03	1.04	1.05	1.06	1.07	1.08	1.09	1.10
2	1.020	1.040	1.061	1.082	1.103	1.124	1.145	1.666	1.188	1.210
3	1.030	1.061	1.093	1.125	1.158	1.191	1.225	1.260	1.295	1.331
4	1.041	1.082	1.126	1.170	1.216	1.262	1.311	1.360	1.411	1.464
5	1.051	1.104	1.159	1.217	1.276	1.338	1.403	1.469	1.539	1.611
6	1.062	1.126	1.194	1.265	1.340	1.419	1.501	1.587	1.677	1.772
7	1.072	1.149	1.230	1.316	1.407	1.504	1.606	1.714	1.828	1.949
8	1.083	1.172	1.267	1.369	1.477	1.594	1.718	1.851	1.992	2.144
9	1.094	1.195	1.305	1.423	1.551	1.689	1.838	1.999	2.171	2.358
10	1.146	1.219	1.344	1.480	1.629	1.791	1.967	2.159	2.367	2.594
11	1.116	1.243	1.384	1.539	1.710	1.898	2.105	2.332	2.580	2.853
12	1.127	1.268	1.426	1.601	1.796	2.012	2.252	2.518	2.813	3.138
13	1.138	1.294	1.469	1.665	1.886	2.133	2.410	2.720	3.066	3.452
14	1.149	1.319	1.513	1.732	1.980	2.261	2.579	2.937	3.342	3.797
15	1.161	1.346	1.558	1.801	2.079	2.397	2.759	3.172	3.642	4.177
16	1.172	1.373	1.605	1.873	2.183	2.540	2.952	3.426	3.970	4.595
17	1.184	1.400	1.653	1.948	2.292	2.693	3.159	3.700	4.328	5.054
18	1.196	1.428	1.702	2.026	2.407	2.854	3.380	3.996	4.717	5.560
19	1.208	1.457	1.754	2.107	2.527	3.026	3.617	4.316	5.142	6.116
20	1.220	1.486	1.806	2.191	2.653	3.207	3.870	4.661	5.604	6.727
21	1.232	1.516	1.860	2.279	2.786	3.400	4.141	5.034	6.109	7.400
22	1.244	1.546	1.916	2.370	2.952	3.604	4.430	5.437	6.659	8.140
23	1.257	1.577	1.974	2.465	3.072	3.820	4.741	5.871	7.258	8.954
24	1.270	1.608	2.033	2.563	3.225	4.049	5.072	6.341	7.911	9.850
25	1.282	1.641	2.094	2.666	3.386	4.292	5.427	6.848	8.623	10.835
26	1.295	1.673	2.157	2.772	3.556	4.549	5.807	7.396	9.399	11.918
27	1.308	1.707	2.221	2.883	3.733	4.822	6.214	7.988	10.245	13.110
28	1.321	1.741	2.288	2.999	3.920	5.112	6.649	8.627	11.17	14.421
29	1.334	1.776	2.357	3.119	4.116	5.418	7.114	9.317	12.17	15.86
30	1.348	1.811	2.427	3.243	4.322	5.743	7.612	10.06	13.27	17.45
31	1.361	1.848	2.500	3.373	4.538	6.088	8.145	10.87	14.46	19.19
32	1.374	1.885	2.575	3.508	4.765	6.453	8.715	11.74	15.76	21.11
33	1.388	1.922	2.652	3.648	5.003	6.841	9.325	12.68	17.18	23.23
34	1.402	1.961	2.732	3.794	5.253	7.251	9.978	13.69	18.73	25.55
35	1.416	2.000	2.814	3.946	5.516	7.686	10.68	14.79	20.41	28.10
36	1.430	2.040	2.898	4.104	5.792	8.147	11.42	15.97	22.25	30.91
37	1.445	2.081	2.985	4.268	6.081	8.636	12.22	17.25	24.25	34.00
38	1.460	2.122	3.075	4.439	6.385	9.154	13.08	18.63	26.44	37.40
39	1.474	2.165	3.167	4.616	6.705	9.704	13.99	20.12	28.82	41.14
40	1.489	2.208	3.262	4.801	7.040	10.29	14.97	21.72	31.41	45.26

Table 15. Values of $\frac{i}{(1+i)^n - 1}$ for Given Values of i and n

n \ i	0.01	0.02	0.03	0.04	0.05	0.06	0.07	0.08	0.09	0.10
1	1.00000	1.00000	1.00000	1.00000	1.00000	1.00000	1.00000	1.00000	1.00000	1.00000
2	0.4975	0.4951	0.4926	0.4902	0.4878	0.4854	0.4838	0.4808	0.4785	0.4762
3	0.3283	0.3268	0.3235	0.3203	0.3172	0.3141	0.3111	0.3080	0.3051	0.3021
4	0.2463	0.2426	0.2390	0.2355	0.2320	0.2286	0.2252	0.2219	0.2187	0.2155
5	0.1960	0.1922	0.1884	0.1846	0.1810	0.1774	0.1739	0.1705	0.1671	0.1638
6	0.1625	0.1585	0.1546	0.1508	0.1470	0.1434	0.1398	0.1363	0.1329	0.1296
7	0.1386	0.1345	0.1305	0.1266	0.1228	0.1191	0.1156	0.1121	0.1087	0.1054
8	0.1207	0.1165	0.1125	0.1085	0.1047	0.1010	0.09747	0.09402	0.09067	0.08744
9	0.1067	0.1025	0.09842	0.09449	0.09069	0.08702	0.08349	0.08008	0.07680	0.07364
10	0.09558	0.09133	0.08723	0.08329	0.07951	0.07587	0.07238	0.06903	0.06582	0.06275
11	0.08645	0.08218	0.07808	0.07415	0.07039	0.06679	0.06336	0.06008	0.05695	0.05396
12	0.07885	0.07456	0.07046	0.06655	0.06283	0.05928	0.05590	0.05270	0.04965	0.04676
13	0.07241	0.06812	0.06403	0.06014	0.05646	0.05296	0.04965	0.04652	0.04357	0.04078
14	0.06690	0.06260	0.05853	0.05467	0.05102	0.04759	0.04435	0.04130	0.03843	0.03575
15	0.06212	0.05783	0.05377	0.04994	0.04634	0.04296	0.03980	0.03683	0.03406	0.03147
16	0.05794	0.05365	0.04961	0.04582	0.04227	0.03895	0.03586	0.03298	0.03030	0.02782
17	0.05426	0.04997	0.04595	0.04220	0.03870	0.03545	0.03243	0.02963	0.02705	0.02466
18	0.05098	0.04670	0.04271	0.03899	0.03555	0.03236	0.02941	0.02670	0.02421	0.02193
19	0.04805	0.04378	0.03981	0.03614	0.03275	0.02962	0.02675	0.02413	0.02173	0.01955
20	0.04542	0.04116	0.03722	0.03358	0.03024	0.02719	0.02439	0.02185	0.01955	0.01746
21	0.04303	0.03879	0.03487	0.03128	0.02800	0.02501	0.02229	0.01983	0.01762	0.01562
22	0.04086	0.03663	0.03275	0.02920	0.02597	0.02305	0.02041	0.01803	0.01590	0.01401
23	0.03889	0.03467	0.03081	0.02731	0.02414	0.02128	0.01871	0.01642	0.01438	0.01257
24	0.03707	0.03287	0.02905	0.02559	0.02247	0.01968	0.01719	0.01498	0.01302	0.01130
25	0.03541	0.03122	0.02743	0.02401	0.02095	0.01823	0.01581	0.01368	0.01181	0.01017
26	0.03387	0.02970	0.02594	0.02257	0.01956	0.01690	0.01456	0.01251	0.01072	0.00916
27	0.03245	0.02829	0.02456	0.02124	0.01829	0.01570	0.01343	0.01145	0.00974	0.00826
28	0.03112	0.02697	0.02329	0.02001	0.01712	0.01459	0.01239	0.01049	0.00885	0.00745
29	0.02990	0.02578	0.02212	0.01888	0.01605	0.01358	0.01145	0.00962	0.00806	0.00673
30	0.02875	0.02465	0.02102	0.01783	0.01505	0.01265	0.01059	0.00883	0.00734	0.00608
31	0.02768	0.02360	0.02000	0.01686	0.01413	0.01179	0.00980	0.00811	0.00669	0.00550
32	0.02667	0.02261	0.01905	0.01595	0.01328	0.01100	0.00907	0.00745	0.00610	0.00497
33	0.02573	0.02169	0.01816	0.01510	0.01249	0.01027	0.00841	0.00685	0.00524	0.00450
34	0.02484	0.02082	0.01732	0.01432	0.01176	0.00960	0.00780	0.00630	0.00508	0.00407
35	0.02400	0.02000	0.01654	0.01358	0.01107	0.00897	0.00723	0.00580	0.00464	0.00369
36	0.02321	0.01923	0.01580	0.01289	0.01043	0.00840	0.00672	0.00535	0.00424	0.00334
37	0.02247	0.01851	0.01511	0.01224	0.00984	0.00786	0.00624	0.00492	0.00387	0.00303
38	0.02176	0.01782	0.01446	0.01163	0.00928	0.00736	0.00580	0.00454	0.00354	0.00275
39	0.02109	0.01717	0.01384	0.01106	0.00877	0.00689	0.00539	0.00419	0.00324	0.00249
40	0.02046	0.01656	0.01326	0.01052	0.00828	0.00646	0.00501	0.00386	0.00296	0.00226

Table 16. Values of $\frac{i}{1 - (1+i)^{-n}}$ for Given Values of i and n

n \ i	0.01	0.02	0.03	0.04	0.05	0.06	0.07	0.08	0.09	0.10
1	1.01000	1.02000	1.03000	1.04000	1.05000	1.06000	1.07000	1.08000	1.09000	1.10000
2	0.5075	0.5151	0.5226	0.5302	0.5378	0.5454	0.5538	0.5608	0.5685	0.5762
3	0.3400	0.3468	0.3535	0.3603	0.3672	0.3741	0.3811	0.3880	0.3951	0.4021
4	0.2563	0.2626	0.2690	0.2755	0.2820	0.2886	0.2952	0.3019	0.3087	0.3155
5	0.2060	0.2122	0.2184	0.2246	0.2310	0.2374	0.2439	0.2505	0.2571	0.2638
6	0.1725	0.1785	0.1846	0.1908	0.1970	0.2034	0.2098	0.2163	0.2229	0.2296
7	0.1486	0.1545	0.1605	0.1666	0.1728	0.1791	0.1856	0.1921	0.1987	0.2054
8	0.1307	0.1365	0.1425	0.1485	0.1547	0.1610	0.1675	0.1740	0.1807	0.1874
9	0.1167	0.1225	0.1284	0.1345	0.1407	0.1470	0.1535	0.1601	0.1668	0.1736
10	0.1056	0.1113	0.1172	0.1233	0.1295	0.1359	0.1424	0.1490	0.1558	0.1628
11	0.09645	0.1022	0.1081	0.1142	0.1204	0.1268	0.1334	0.1401	0.1470	0.1540
12	0.08885	0.09456	0.1005	0.1066	0.1128	0.1193	0.1259	0.1327	0.1397	0.1468
13	0.08241	0.08812	0.09403	0.1001	0.1065	0.1130	0.1197	0.1265	0.1336	0.1408
14	0.07690	0.08260	0.08853	0.09467	0.1010	0.1076	0.1144	0.1213	0.1284	0.1358
15	0.07212	0.07783	0.08377	0.08994	0.09634	0.1030	0.1098	0.1168	0.1241	0.1315
16	0.06794	0.07365	0.07961	0.08582	0.09227	0.09895	0.1059	0.1130	0.1203	0.1278
17	0.06426	0.06997	0.07595	0.08220	0.08870	0.09545	0.1024	0.1096	0.1171	0.1247
18	0.06098	0.06670	0.07271	0.07899	0.08555	0.09236	0.09941	0.1067	0.1142	0.1219
19	0.05805	0.06378	0.06981	0.07614	0.08275	0.08962	0.09675	0.1041	0.1117	0.1196
20	0.05542	0.06116	0.06722	0.07358	0.08024	0.08719	0.09439	0.1019	0.1096	0.1175
21	0.05303	0.05879	0.06487	0.07128	0.07800	0.08501	0.09229	0.09983	0.1076	0.1156
22	0.05086	0.05663	0.06275	0.06920	0.07597	0.08305	0.09041	0.09803	0.1059	0.1140
23	0.04886	0.05467	0.06081	0.06731	0.07414	0.08128	0.08871	0.09642	0.1044	0.1126
24	0.04707	0.05287	0.05905	0.06559	0.07247	0.07968	0.08719	0.09498	0.1030	0.1113
25	0.04541	0.05122	0.05743	0.06401	0.07095	0.07823	0.08581	0.09368	0.1018	0.1102
26	0.04387	0.04970	0.05595	0.06257	0.06956	0.07690	0.08456	0.09251	0.1007	0.1092
27	0.04245	0.04829	0.05456	0.06124	0.06829	0.07570	0.08343	0.09145	0.09974	0.1083
28	0.04112	0.04697	0.05329	0.06001	0.06712	0.07459	0.08239	0.09049	0.09885	0.1075
29	0.03990	0.04578	0.05212	0.05888	0.06605	0.07358	0.08145	0.08962	0.09806	0.1067
30	0.03875	0.04465	0.05102	0.05783	0.06505	0.07265	0.08059	0.08883	0.09734	0.1061
31	0.03768	0.04360	0.05000	0.05686	0.06413	0.07179	0.07980	0.08811	0.09669	0.1055
32	0.03667	0.04261	0.04905	0.05595	0.06328	0.07100	0.07907	0.08745	0.09610	0.1050
33	0.03573	0.04169	0.04816	0.05510	0.06249	0.07027	0.07841	0.08685	0.09524	0.1045
34	0.03484	0.04082	0.04732	0.05432	0.06176	0.06960	0.07780	0.08630	0.09508	0.1041
35	0.03400	0.04000	0.04654	0.05358	0.06107	0.06897	0.07723	0.08580	0.09464	0.1037
36	0.03321	0.03923	0.04580	0.05289	0.06043	0.06840	0.07672	0.08535	0.09424	0.1033
37	0.03247	0.03851	0.04511	0.05224	0.05984	0.06786	0.07624	0.08492	0.09387	0.1030
38	0.03176	0.03782	0.04446	0.05163	0.05928	0.06736	0.07580	0.08454	0.09354	0.1028
39	0.03109	0.03717	0.04384	0.05106	0.05887	0.06689	0.07539	0.08419	0.09324	0.1025
40	0.03046	0.03656	0.04326	0.05052	0.05828	0.06646	0.07501	0.08386	0.09296	0.1023

Table 17. Experimental and Commercial Nuclear Power Plants[a]

Organization	Atomic Energy Commission				Pacific Gas & Elec. Co. and General Electric Co.	Duquesne Light Company and Atomic Energy Comm.	U.S. Army Corps of Engineers
Plant name	Experimental breeder reactor No. 1 (EBR-1)	Boiling reactor experiments 2, 3, 4 (BORAX)	Experimental boiling-water reactor (EBWR)	Sodium reactor experiment (SRE)	Vallecitos boiling-water reactor	Shippingport, Pa. (PWR)	Army package power reactor (APPR-1)
Plant location	NRTS, Idaho	NRTS, Idaho	Lemont, Ill.	Santa Susana Mts., Calif.	Pleasanton, Calif.	Shippingport, Pa.	Fort Belvoir, Va.
Date first critical	1951	1954	Dec. 1, 1956	Apr. 25, 1957	Aug. 5, 1957	Dec. 2, 1957	Apr. 8, 1957
Date first full power	1951	1954	Dec. 29, 1956	July, 1957	Oct. 24, 1957	Dec. 23, 1957	Apr. 15, 1957
Reactor type	Fast breeder	Boiling water	Boiling water	Sodium graphite	Dual-cycle BW[d]	Pressurized water	Pressurized water
Reactor builder	Argonne Natl. Lab.	Argonne Natl. Lab.	Argonne Natl. Lab.	Atomics International[c]	General Electric Co.	Westinghouse Elec. Corp.	Alco Products, Inc.
Reactor thermal power, mwt	1.40	15.5	20 des.; 61.7 actual	20	20 des.; 30 actual	236	10
Plant electrical capacity, mwe	0.15	2.0	6.25 capability	6.0 (7.5-mw turbine)	5.0	68 (100-mw turbine)	1.975 (2.2-mw turbine)
Reactor cost (latest estimate)	……	……	……	$7,300,000	$2,500,000[e,f]	$59,200,000	
Turbo-generator cost	……	……	……	$1,500,000	$ 572,000	$18,300,000	
Total plant cost	$2,732,000	……	$4,628,000	$8,800,000	$3,072,000	$77,500,000	$4,300,000
Unit plant investment, dollars per kw	……	……	740	1,170 @ 7.5 mw	614 @ 5 mw	1290 @ 60-mw net	2150
Unit energy cost, mills per kwhr	……	……	52 (@ 80% use factor)[b]	……	……	64	
Moderator	None	Light water	Light water	Graphite	Light water	Light water	Light water
Coolant	Sodium potassium	Light water	Light water	Sodium	Light water	Light water	Light water
Coolant temperature, °F	442–600	419	488	500–960	545	510–540	431–450
Coolant pressure, psig	15	300	600	0	1,000	1,985	1,185
Steam temperature, °F	525 (214-F feedwater)	415	479 (110-F feedwater)	825	545	Saturated (344-F feed)	407 (full load)
Steam pressure, psig	400	285	560	600	1,000	870 no-load; 585 full	407 no-load; 185 full
Steam flow, lb per hr	3,630	……	60,600 design	60,000	90,000 @ 30 mwt	861,000	34,070
Fuel type	90% U-235 in seed Natural-U blanket	Slightly enriched uranium	1.4% U-235	2.8% U-235	Fully enriched uranium	90% U-235 in seed Natural-U blanket	Highly enriched UO_2
Engineers	Argonne Natl. Lab.	Argonne Natl. Lab.	Sargent & Lundy	Atomics International	Bechtel Corp.	Stone & Webster Engrg. Corp. Burns and Roe, Inc.	Stone & Webster Engrg. Corp.

Table 17. Experimental and Commercial Nuclear Power Plants (*Continued*)

Organization	Consolidated Edison Company of New York, Inc.	Commonwealth Edison Co. and Nuclear Power Group, Inc.[i]	Yankee Atomic Elec. Co.	Power Reactor Development Company and Detroit Edison Co.	Consumers Public Power District	Northern States[k] Power Company	Pacific Gas & Electric Company
Plant name	Indian Point Station	Dresden Nuclear Power Station	Yankee Pressurized Water Power Plant	Enrico Fermi Atomic Power Plant	Sodium Graphite Power Plant	Pathfinder	Humboldt Bay
Plant location	Buchanan, N.Y.	Morris, Ill.	Rowe, Mass.	Lake Erie, nr. Detroit	Beatrice, Nebr.	Sioux Falls, S. Dak.	Eureka, Calif.
Date first critical	1960	1960	1960	1960	1962	1962	1962
Date first full power	1960	1960	1960	1960	1962	1962	1962
Reactor type	Pressurized water[h]	Dual-cycle boiling water	Pressurized water	Fast breeder	Sodium graphite	Boiling water[l]	Boiling water
Reactor builder	Babcock and Wilcox Co.	General Electric Co.	Westinghouse Elec. Corp.	PRDC[j]	Atomics Internat'l.	Allis-Chalmers Mfg. Co.	General Electric Co.
Reactor thermal power, mwt	585	627.5	492	300	245	203	171
Plant electrical capacity, mwe	275[f]	180	134 (145-mw turbine)	100	75	62 (66-mw turbine)	50–60
Reactor cost (latest estimate)	$38,000,000	……	$23,600,000	$34,200,000	$29,000,000		
Turbine-generator cost	$45,000,000[g]	……	……	……	$11,300,000		
Total plant cost	$90,000,000	$45,000,000	$57,000,000	$67,146,000	$40,300,000	$28,800,000	$20,000,000
Unit plant investment, dollars per kw	327	250	425	671	537	465	333 (@ 60 mw)
Unit energy cost, mills per kwhr	10–12	7.5 (excl. $15 million R&D)	14–16	……	10–11	……	8.7–1st core
Moderator	Light water	Light water	Light water	None	Graphite	Light water	Light water
Coolant	Light water	Light water	Light water	Sodium	Sodium	Light water	Light water
Coolant temperature, °F	480–510	547	487–535	550–800	500–925	489 (blg), 825 (superheating)	545
Coolant pressure, psig	1,485	1,000	1,985	0	46	600	1,000
Steam temperature, °F	1,000 (332-F feed)	541 primary	471	755 (400-F feed)	825 (300-F feed)	825 (300-F feed)	545
Steam pressure, psig	355	965 prim.; 460 sec.	505	600	800	600	1,000
Steam flow, lb per hr	1,900,000	1.4×10^6 prim.; 1.2×10^6 sec.	1,800,000	1,000,000	730,000	600,000	580,000 (@ 50 mw)
Fuel type	Highly enriched UO_2 ThO_2 blanket	1.5% U-235 in UO_2	2.6% U-235 Uranium oxide compacts	25% U-235 with depleted-U blanket	2.5% U-235	1.6% U-235 for boiling 20% U-235 for superheating	2% U-235
Engineers	Vitro Engineering Co.	Bechtel Corp.	Stone and Webster Engineering Corp.	Commonwealth Assoc. Inc. United Engrs. & Const., Inc.	Bechtel Corp.	Pioneer Service and Engineering Co.	Bechtel Corp.

TABLE 17. EXPERIMENTAL AND COMMERCIAL NUCLEAR POWER PLANTS (*Continued*)

Organization	General Public Utilities	Florida West Coast and East Central Nuclear Groups[m]	Rural Cooperative Power Association	Carolinas-Virginia Nuclear Power Associates, Inc.	City of Piqua, Ohio	Consumers Power Company	Atomic Energy Commission
Plant name	Saxton	...	Elk River boiling reactor	...	Municipal plant	Big Rock Point	Experimental breeder reactor No. 2 (EBR-2)
Plant location	Saxton, Pa.	Florida west coast	Elk River, Minn.	Parr Shoals, S.C.	Piqua, Ohio	Big Rock Point, Mich.	NRTS, Idaho
Date first critical	1962	1963	1960	1962	1961	1962	1960
Date first full power	1962	1963	1960	1962	1961	1962	1960
Reactor type	Pressurized water	Gas cooled, D_2O moderated	Boiling water[n]	PWR, pressure tube	Organic moderated	Boiling water-forced circulation	Fast, enriched U-Pu
Reactor builder	Westinghouse Elec. Corp.	General Nuclear Engrg. Corp.	ACF Industries, Inc.	Westinghouse Elec. Corp.	Atomics Internatl. Div.	General Electric Co.	Argonne Nat. Lab.
Reactor thermal power, mwt	20	173	73	60.5	45.5	...	62.5
Plant electrical capacity, mwe	12	46 (50-mw turbine)	22	17.0	11.4	50–75	20
Reactor cost (latest estimate)	$6,300,000	...	...	...	$12,300,000	...	
Turbine-generator cost	Existing turbine	...	$1,800,000	...	$4,000,000	...	
Total plant cost	...	$34,900,000	$13,300,000	$36,700,000	$16,300,000	...	$29,100,000
Unit plant investment, dollars per kw	...	759	605	2,160	1,430	...	1,455
Moderator	Light water	Heavy water (D_2O)	Light water	Heavy water	Hydrocarbon	Light water	None
Coolant	Light water	Gas	Light water	Heavy water	Hydrocarbon	Light water	Sodium
Coolant temperature, °F	505 average	1020	528 water, 533 steam	550 average	617 average	593–	720–900
Coolant pressure, psig	1185	520	900	1500	50	1000–1450	100
Steam temperature, °F	Saturated	1000	825	725	550	593–	850
Steam pressure, psig	300–450 (no load)	1,000–1,800	600	400	450	1000–	1,250
Steam flow, lb per hr	203,000	...	225,000	203,000	150,500	...	200,000
Fuel type	Slightly enriched UO_2 pellets	Slightly enriched U	9.8% enriched UO_2 ThO_2 fertile material	2% U-235	Slightly enriched U	Slightly enriched UO_2 pellets	60% enriched U
Engineers	Gilbert Associates	General Nuclear Engrg. Corp.	...	Stone and Webster Engrg. Corp.	...	Bechtel Corp.	

TABLE 17. EXPERIMENTAL AND COMMERCIAL NUCLEAR POWER PLANTS (*Continued*)

Organization	Atomic Energy Commission				Philadelphia Elec. Co., HTRDA and AEC	U.S. Army Corps of Engineers	Michigan Chemical Corporation
Plant name	Homogeneous reactor experiment No. 2 (HRE-2)	Organic moderated reactor experiment (OMRE)	Los Alamos power reactor exp. (LAPRE-2)	Liquid-metal fuel reactor experiment (LMFRE)	Peach Bottom	Army package-power reactor (APPR-1A)	Process steam reactor (proposed)
Plant location	Oak Ridge, Tenn.	NRTS, Idaho	Los Alamos, N. Mex.	Not announced	Peach Bottom, Pa.	Nr. Fairbanks, Alaska	St. Louis, Mich.
Date first critical	1957—delayed by corrosion problems	1957	1958	...	1963	1960	Not announced
Date first full power	...	1957	1958	...	1963	1960	Not announced
Reactor type	Enriched U, homogeneous	Organic moderated	Aqueous homogeneous	Liquid-metal fuel	Gas cooled	Pressurized water	Pressurized water
Reactor builder	Oak Ridge Natl. Lab.	Atomics Internatl. Div.	Los Alamos Scien. Lab.	Babcock & Wilcox Co.	General Dynamics Corp.	Alco Products, Inc.	Martin Co.
Reactor thermal power, mwt	5.0	16	1.0	20	115	20 approx.	45
Plant electrical capacity, mwe	0.3 (existing generator)	0	0	0	30–40	1.7 (10-mw heating)	None
Reactor cost (latest estimate)	...	$2,531,000	$100,000	$15,000,000	...	...	$2,500,000
Turbine-generator cost	...	...	...	...	...	...	0
Total plant cost	$3,833,000	...	...	...	$610,000,000	...	$2,836,000
Unit energy cost, mills per kwhr	...	...	...	...	...	...	77.4 cents per 1,000 lb steam
Moderator	Heavy water	Diphenyl	Light water	Graphite	Graphite	Light water	Light water
Coolant	Fuel solution	Diphenyl	Light water	Fuel solution	Helium	Light water	Light water
Coolant temperature, °F	493–572	500–700	600 out	825–925	1380 average		
Coolant pressure, psig	1750–2000	300	800	...	300		
Steam temperature, °F	471	None	492	...	1000	...	366
Steam pressure, psig	505	None	600	...	1450	...	150
Steam flow, lb per hr	...	None	...	...	367,000	...	150,000
Fuel type	90% enriched U-235 in UO_2SO_4-D_2O	90% U-235	90% U-235	90% U-235	Enriched UC_2 + ThC_2 in graphite	Highly enriched U	Slightly enriched U
Engineers	...	Atomics Internatl. Div.	...	...	Bechtel Corp.	Peter Kiewit Sons Co. of Omaha	H. K. Ferguson Co.

[a] Courtesy of *Power*.

[b] With 15-mw turbine generator, costs estimated at 32 mills per kilowatthour

[c] Southern California Edison Co. supplies turbine generator and buys energy.

[d] Boiling-water reactor can work as natural or forced circulation and single or dual cycle by venting flashed steam to atmosphere.

[e] Excluding land and miscellaneous items needed for a commercial plant.

[f] 112 mwe supplied by an oil-fired superheater.

[g] Includes $13,000,000 for an oil-fired superheater.

[h] Converter changing Th-232 to U-233.

[i] Includes American Electric Power Corp., Bechtel Corp., Central Illinois Lighting Co., Commonwealth Edison Co., Illinois Power Co., Kansas City Power & Light Co., Pacific Gas & Electric Co., Union Electric Co.

[j] Power Reactor Development Company consisting of 6 manufacturers and 18 electrical utilities.

[k] With Central Atomic Power Associates consisting of 11 electrical utility companies.

[l] Controlled water circulation and nuclear-heated superheater.

[m] Florida West Coast and East Central consists of two Florida and 12 northern utility companies.

[n] Closed cycle; oil-fired superheater for 7 mwe.

APDA, Atomic Power Development Associates, Inc.
NRTS, National Reactor Testing Station.
R & D, Research and Development.
HTRDA, High Temperature Reactor Development Associates (41 utilities).

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